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U. S. S. *HOLLAND.*

DESCRIPTION AND TRIALS.

BY COMMANDER A. M. CHARLTON, U.S.N., MEMBER.

The U. S. S. *Holland,* Submarine Tender Number 3, was commissioned on June 1, 1926, at the Navy Yard, Puget Sound, Washington, where she was built.

Just as the size of submarines has increased, so has the size of the *Holland* been increased over previous submarine tenders in our Navy. The *Fulton,* Submarine Tender Number 1, completed in 1914, is a vessel 226 feet long, displacing about 1400 tons and on her full power trial made a speed of 12.34 knots. She is a single screw vessel driven by a Diesel engine which developed 1097 I.H.P. on her trials. The *Bushnell,* Submarine Tender Number 2, completed in 1915, is 350 feet long, displacing 3575 tons and designed for a speed of 14 knots. On her trials she developed

2617 S.H.P. and made a speed of 14.15 knots. The *Bushnell* is equipped with two oil fired Yarrow boilers and a Parsons geared turbine driving a single screw.

The *Holland* was named after John Holland, an inventor who first laid his plans for a submarine before the Navy Department in 1875. He received the first contract from the United States for a submarine in 1895. This vessel, the original *Plunger,* was never finished, but the amounts paid toward its construction were credited on a new contract in 1900, covering two submarines. These vessels, the *Holland* and *Plunger,* were the first submarines on the Navy list.

The *Holland* is the third tender recently completed in our Navy Yards. The *Dobbin* and *Whitney,* Destroyer Tenders Numbers 3 and 4, were recently completed at the Philadelphia and Boston Navy Yards respectively.

Although the *Holland* is practically a sister ship of the *Dobbin* and *Whitney* as far as displacement, propelling equipment, speed and armament are concerned, she differs in a number of ways, because of the difference in her mission.

Submarines come alongside a tender primarily for repairs and to replenish their supplies. Inasmuch, however, as living conditions on a submarine are uncomfortable at best, arrangements are made on the *Holland* to accommodate the officers and crews of eight fleet submarines. Such accommodations are unnecessary on destroyer tenders and are not fitted on the *Dobbin* and *Whitney.*

A submarine tender should be able to lift partially damaged submarines. To accomplish this the *Holland* has a clipper bow extending thirty feet forward of the forward perpendicular. This bow is built up internally with very strong bracing so that it can be used as a crane. Sheaves are fitted on the bow so that the bower anchor chains may be put around the damaged submarine and the windlass engine used to lift. The effect can be added to by flooding the peak tanks before the lifting chains are put in place, and pumping them out to gain additional lift.

Submarine storage batteries must be charged as they run down. To avoid running the submarine's motors while alongside the tender, battery charging generators are installed on the *Holland* for this purpose.

Very pure distilled water is needed in storage batteries, much purer than the usual output from the ship's distilling plant. A re-evaporating plant is therefore installed on the *Holland,* which takes its water from the ship's evaporators and re-distills it for storage battery use.

Inasmuch as the *Dobbin* and *Whitney* were fully described in the JOURNAL for August, 1925, this article will be confined principally to pointing out the differences in the *Holland* from those two vessels. Where no mention is made herein of specific apparatus it may be taken as similar to that on the *Dobbin* and *Whitney.*

The keel of the *Holland* was laid on 11 April, 1921. She was launched by flooding the dock wherein she was built on 12 April, 1926.

Her beam, draft and displacement are the same as those of the *Dobbin,* but due to the clipper bow previously mentioned she is 513 feet in length overall instead of 483 feet, 9½ inches.

The *bridge deck* of the *Holland* is given over entirely to quarters and offices with the exception of a small radio transmitting room on the starboard side aft.

The *Holland's* superstructure deck is very much the same as the *Dobbin's* except for the arrangement of boat stowage.

The five-inch guns which are mounted aft on this deck on the *Dobbin* are moved farther forward on the *Holland.*

The arrangement of the *main* and *second decks* on the two vessels varies somewhat due to the necessity for finding berthing space for submarine officers and crews. For this reason the shops on the *Holland* are concentrated around the machinery hatch on the main deck, while on the *Dobbin* considerable space on the second deck is used for shops.

The arrangement of the *lower* decks is in general the same on the two vessels.

COMPLEMENT.

The complement of the *Holland* is 32 officers and 334 men as compared with 30 ship's officers, 8 staff officers and 416 men on the *Dobbin.*

BOATS.

The *Holland* carries the following boats:

4—40-foot motor launches
2—24-foot motor launches
4—35-foot motor boats
2—30-foot whale boats
2—16-foot dinghies
1—40 foot admiral's barge

These boats have a carrying capacity of 643 men as compared to the *Dobbin's* boat capacity of 1024 men.

ANCHOR HANDLING GEAR.

The anchor windlass was manufactured by the American Clay Machinery Company. It is of the horizontal spur gear type driven by a two-cylinder steam engine, 13-inch diameter cylinders and 13-inch stroke. It has two speeds and is reversible. It is capable of raising two 14,000-pound anchors simultaneously at the rate of 6 fathoms per minute from a depth of over 60 fathoms. When operated for lifting submarines, it will develop a total pull of 650,000 pounds upon 260 fathoms of 2¾-inch Navy type stud link chain. A special cast steel chain suitable for this service is to be provided.

STEERING GEAR AND DECK WINCHES.

The steering gear was furnished by the Lidgerwood Mfg. Company. The engine has two 10-inch by 10-inch cylinders. It is controlled by means of wire ropes from the steering stations.

The seven deck winches were manufactured by the Lidgerwood Mfg. Company. They are arranged as on the *Dobbin* and *Whitney*.

RADIO AND SOUND INSTALLATION.

The radio equipment is similar to that on the *Dobbin*.

There is a keel oscillator, submarine type, installed. A sound receiver and a sonic depth finder are to be installed.

ENGINEER DEPARTMENT.

BOILERS.

The arrangement of fire rooms and the boilers themselves are similar to those installed on the *Dobbin* and *Whitney*. Three Mayflower-Lodi blower burners as manufactured by the Babcock and Wilcox Co. are installed on each boiler, which will allow the vessel to be operated with open fireroom up to a speed of about 14 knots. For speeds above this up to full power the firerooms are closed and the forced draft blowers started. The boilers are insulated in accordance with the new General Specifications for Machinery dated October 1, 1925, which call for 2 inches of diatomaceous earth on the tube casings, 4½ inches of fire brick backed with 4½ inches of diatomaceous earth on front, back and side walls, and 2½ inches of fire brick with 4½ inches of diatomaceous earth under it on the furnace bottoms. Soot blowers are to be installed in each boiler, the installation consisting of an element over each water drum, fitted to rotate and blow soot from the tube bank. Safety valves are set to lift 220 pounds gauge.

There are two forced draft blowers in each fire-room. These are vertical, turbine-driven, Terry type GV, with 31-inch semi-silent fans. Each blower has a capacity of 25,000 cubic feet per minute against 7 inches air pressure.

The fuel oil system consists of a number of fuel oil tanks, heated, when necessary, by steam coils, from which oil is taken by the fuel oil service pumps and sent to the burners through fuel oil heaters.

There are four fuel oil pumps, two in each fire-room. Three of the pumps are 5 by 4 by 8 inch, and one 5 by 3½ by 8 inch, Worthington. There is also a fuel oil supply pump 7½ by 8½ by 8 inches in the forward boiler room, which may be used as a booster pump if necessary. There is one Reilly No. 4 fuel oil heater in each fireroom.

Feed water is supplied to the boilers by two 10 by 7 by 12 inch, Worthington, main feed pumps. The water passes through a No. 47 Reilly multicoil feed heater on its way to the boilers. The auxiliary exhaust is used to heat the feed water. There is an auxiliary feed pump in each fireroom, similar to the main feed pumps.

The main propelling turbines were manufactured by the Navy Yard, Puget Sound, from plans and specifications purchased from the Parsons Marine Steam Turbine Company, which are the same as those used for the *Dobbin* and *Whitney*. They were designed for 7000 S.H.P. at a speed of a 105 R.P.M. on the propeller shaft. The reduction gears were cut and machined in the Navy Yard, Philadelphia. The designed speed of the vessel was 16 knots. The propeller was manufactured in the Navy Yard, Puget Sound, from plans prepared in the Bureau of Engineering.

The main condenser was manufactured by the Navy Yard, Puget Sound. It is pear-shaped in cross section and has a cooling surface of 8201 square feet. There are 4177 ⅝-inch tubes twelve feet long between tube sheets. The tubes are expanded on one end and packed with John Crane metallic packing on the other.

The main circulating pump and engine were manufactured by the Navy Yard, Puget Sound, from plans prepared in the Bureau of Engineering. The engine is single cylinder, 11-inch diameter, with 12-inch stroke. The pump is centrifugal and has a capacity of 10,000 gallons per minute at 250 R.P.M.

Water is removed from the condenser by means of a turbine driven centrigugal condensate pump manufactured by the C. H. Wheeler Mfg. Co. It has a capacity of 175 gallons per minute.

Air is removed from the condenser by means of two type G-3 radojet air ejectors manufactured by the C. H. Wheeler Mfg. Co. They are designed to maintain 28 inches of vacuum with 60 degrees F. circulating water.

As the main condenser is above the bottom of the low pressure turbine casing, a Wilmon vacuum trap is installed to drain the turbine casing. A turbine drain pump is also installed as a standby. This is a 6 by 10 by 8 inch Worthington featherweight air pump.

The turbine bearings and reduction gear are lubricated by forced lubrication system. Oil is supplied by three 8 by 10 by 12 inch Worthington pumps. Oil is cooled in two Reilly No. 116 multi-whirl coolers of 300 square feet cooling surface each. They are designed to keep the lubricating oil down to 100 degrees F. with 60 degrees F. cooling water. The coolers are supplied water by two Worthington 8 by 9 by 12 inch pumps.

The electric generators for ship's light and power consist of two 200 K.W. and one 100 K.W. sets installed, as on the *Dobbin*

and *Whitney,* on a dynamo flat in the engine room. The generators are of General Electric make, compound-wound, and generate current at 125 volts. They are driven by Moore turbines geared down from 4500 R.P.M. to 1200 R.P.M. on the generators.

The condenser has a cooling surface of 2387 square feet. Circulating water is provided by a 14-inch turbine driven centrifugal pump. This pump was manufactured in the Navy Yard, Puget Sound. The condensate is removed by means of a Worthington centrifugal pump driven by a Terry turbine. Air is removed by two type D-2 Wheeler radojet air ejectors.

The distribution of light and power, and the arrangement of interior communication circuits is quite similar to those on the *Dobbin* and *Whitney,* the differences being due to the general arrangements of the vessels. The ship's service telephone system on the *Holland* is of the automatic type, with twenty-nine telephones. When alongside a dock shore calls must be made by a switchboard operator.

To avoid the running of a submarine's main generators for charging its storage batteries while alongside the tender, two 400 K.W. generators are installed in a dynamo room on the platform deck just forward of the boilers. These machines were manufactured by the General Electric Co. They are turbine driven with reduction gears, the turbines running at 5000 R.P.M., and the generators at 1000 R.P.M. The generators are shunt wound and the voltage produced can be varied from 325 to 125. There is a battery charging switchboard in this room, from which leads are run to the ship's side for charging submarine batteries in place. Twenty-four hundred feet of 800,000 c.m. cable are provided for making the temporary connections. A motor generator to provide the very low voltage necessary in charging a single cell is to be installed.

The condenser has a cooling surface of 2387 square feet. Condensing water is supplied by a turbine driven centrifugal pump. There is a turbine driven condensate pump discharging to a hotwell. Air is removed by two Wheeler radojet air ejectors. The vacuum equipment is similar to that on ship's main plant. A 4½ by 6 by 6 inch hotwell pump discharges to the main feed tank.

DISTILLING PLANT.

The distilling plant is similar to that of the *Whitney*, two quadruple effect units with electric driven auxiliaries being installed. The plant has a rated capacity of 40,000 gallons per day with a 30 per cent. overload capacity when clean.

The water produced by the main distilling plant is not sufficiently pure for use in submarine storage batteries. To provide water of the necessary purity, a re-distilling plant is installed above the main distilling plant. This plant is a high pressure single effect evaporator using steam at 70 pounds pressure. Feed is supplied by one of the main plant fresh water pumps which also is used as a distiller circulating pump. Fresh water is used as condensing water for the distiller, the supply coming from a reserve feed tank. A 3 by 1¾ by 4 inch pump puts the distilled water into the battery water tank. From here water flows by gravity to connections on the ship's side and to the submarine battery workshop.

The evaporator is of the Reilly multi-coil type with a capacity of 2700 gallons per day.

The distilling condenser is of the Bureau standard type with a capacity of 2500 gallons per day.

The re-distilling plant is designed to produce water of not over .1 grain salinity.

REFRIGERATING PLANT.

The cold storage capacity is similar to the *Dobbin*. There are two CO_2 compressors manufactured by the American Carbonic Machinery Company, each of 8 tons' capacity. Either will maintain the plant. A shell type condenser is used, its circulating water being supplied by two 7 by 6 by 12 inch Cameron pumps. Brine is circulated to the cold storage rooms by means of two Worthington 7 by 7 by 12 inch pumps.

AIR COMPRESSOR PLANT.

The installation for shops, guns, boiler cleaning, etc., is the same as on the *Dobbin*. The galley air compressor has a capacity of 50 free air per minute compressing to 40 pounds pressure.

SHOPS.

The shop equipment on the *Holland* is not quite so extensive as that on the *Dobbin* and *Whitney,* due to the difference in the class of work handled.

The following machine tools are installed; in addition to complete outfits of hand tools:

MACHINE SHOP.

1—Electric welder. Lincoln Electric Co.

1—16-inch by 32-inch by 10 feet gap lathe. Rann Larmon Co.

2—14-inch by 8 feet engine lathes. Graves Klusman Co.

2—8½-inch by 22-inch precision lathes. Rivett Lathe and Grinder Co.

1—Universal Milling Machine. Cincinnati Milling Machine Co.

1—16-inch crank shaper. Smith and Miller Co.

1—8-inch power hack saw. Robertson Foundry and Machinery Co.

1—Grinder, 6-inch wheel. Cincinnati Electric Tool Co.

1—Drill press, 20-inch. Rockford Machine and Tool Co.

1—30-inch radial drill table 28 inches by 36 inches. Fosdick Machine Tool Co.

1—16-inch sensitive drill, floor type. James Clark, Jr. Electric Co.

1—12-inch by 2 inch emery grinder. James Clark, Jr. Electric Co.

1—Boring mill, horizontal, 54-inch diameter by 60-inch. Universal Boring Machine Co.

4—6-inch machinist's vises.

1—Universal grinder No. 2. Brown and Sharp Mfg. Co.

1—Face plate, 5 feet by 5 feet. Puget Sound Navy Yard.

1—Lathe, engine, 16 inches by 6 feet. Monarch Machine Tool Co.

A core oven, arbor press and 30-ton forcing press are to be installed in the machine shop.

PATTERN SHOP.

1—24-inch by 6 feet patternmaker's lathe. Oliver Machinery Co.
1—33-inch band saw. Fay and Eagen Co.
1—Universal wood trimmer. Oliver Machinery Co.
2—Electric glue pots.
1—24-inch by 3-inch grindstone.
1—Patternmaker's vise.
1—Circle rip saw.

PIPE AND COPPER SHOP.

1—Fuel oil forge. Hauck Mfg. Co.
1—Hand shear.
1—Pipe bending machine. American Pipe Bending Machine Co.
1—Combination vise.

BLACKSMITH SHOP.

1—Fuel oil forge. Mircs Fuel Oil Equipment Co.
1—Hand forge.
1—100-pound steam hammer.
1—200-pound anvil.
2—5-inch blacksmith vises.

FOUNDRY.

1—Furnace, crucible, oil burning 165 pound capacity. Mircs Fuel Oil Equipment Co.

ELECTRICAL SHOP.

1—Varnish sprayer.
1—Armature coil winding lathe, 34 inches by 5½ feet. Peerless Equipment Co.
1—Drill press, 20-inch. Independent Machine Co.
1—Bench drill.
1—Precision lathe, 8½-inch by 22-inch. Rivett Lathe and Grinder Co.
1—Coil winding machine. Universal Winding Co.
1—Bench grinder, 12-inch by 2-inch. Cincinnati Electric Tool Co.
1—Paper cutter, 40-inch blade. John Jacques Co.

1—Armature coil taping machine. Peerless Equipment Co.
1—Armature baking oven, 5 feet by 3¼ inches by 2 feet 4½ inches.
1—Armature baking oven, 2 feet by 2 feet 11¾ inches by 2 feet.

STORAGE BATTERY REPAIR SHOP.

This room has its deck covered with sheet lead which is flashed up the sides. The interior is painted with acid-resisting paint. It is equipped with a vulcanizing set, a bench type lead burning outfit (electric), as well as acetylene burners.

TORPEDO REPAIR SHOP.

A spacious torpedo repair shop is well equipped with machine tools and testing stands for making repairs, adjusting and testing torpedoes.

OPTICAL SHOP.

1—Universal milling machine. Brown and Sharp Mfg. Co.
1—Portable sensitive bench drill. Clark Electric Drill Co.
1—Bench buffing lathe machine. Cincinnati Electric Co.
1—Two-wheel emery grinder. Hisey Wool Machine Co.
1—Telescope collimeter.
1—Electric plating outfit.
1—Electric disc stove.
1—Electric soldering iron.

TRIALS OF THE "HOLLAND."

The standardization trials of the *Holland* were run on the measured mile off Vashon Island, Puget Sound, on 16 August, 1926. The results of these runs were as follows:

Speed Knots	R. P. M.	Shaft Horsepower
7.986	52.02	972
10.299	66.46	1876.65
12.384	79.80	3123.86
14.228	92.33	4868.19
15.832	105.77	7422.92

The estimated displacement on these trials was 11,038 tons. The *Holland* had been docked 20-30 July, 1926, so that at the time of these trials the bottom was nearly clean.

Following the standardization trials, fuel economy trials at various speeds were run, during which the following data were taken.

Run	Full Power	14 knots	12 knots	10 knots
Date, August	18	18	19	17
Duration, hours	4	4	4	4
R. P. M.	106.06	92.04	78.48	65.04
Knots	15.87	14.09	12.18	10.08
Displacement	10,995	11,005	10,936	11,022
Shaft horsepower	7005	4714	2819	1524
Number of boilers	2	2	2	2
Burners per boiler	3	3	3	3
Gallons fuel per hour (by soundings)	981.25	718.75	481.25	312.5
Gravity Baumé	17.8	17.8	17.8	17.8
Temperature oil to heater, degrees F.	75	77	79	82
Pounds fuel per hour	7708	5631	3768	2445
Pounds fuel per knot	485.6	400	309.3	242.5
Pounds fuel per S. H. P.	1.098	1.194	1.336	1.605
Air pressure inches water	2.7	Open fireroom	Open fireroom	Open fireroom

While the shaft horsepowers tabulated above are at variance with those recorded on the standardization runs, the Trial Board considered the former the more accurate.

Blower burners with open fireroom were used on all economy trials except full power. The temperatures recorded in the firerooms were quite low, the average in front of the boilers at 12 knots being 120 degrees F.

The operation of the main machinery and auxiliaries during the trial was quite satisfactory.

REDUCTION GEARS IN U. S. NAVAL VESSELS.

By Howard J. Ball, Civil Member.

When the steam turbine reached a stage of development where it became a recognized rival of the reciprocating engine as a means of ship propulsion, it soon became apparent that, due to the high rotational speed of the turbine and the necessarily low speed of the propeller, some reliable means must be introduced whereby the power developed by the turbine might be efficiently transmitted to the propeller without a material sacrifice in the speed and efficiency of either.

Speed reduction has been more or less successfully accomplished by electrical and hydraulic means, and, though both electric and hydraulic transmission were primarily thought to have greater possibilities than mechanical gearing, the problem was finally solved by the development of the mechanical reduction gear, and, as the direct connected turbine superseded the reciprocating engine, so the geared turbine gradually replaced the direct type of turbine drive. This radical change in ship propulsion was made, for the greater part, during the late war period.

The present speeds for main propelling turbines in the Navy range up to 6000 R. P. M., with propeller speeds ranging from 450 R. P. M. for destroyers and fast cruisers, down to 90 R. P. M. for auxiliary vessels, such as transports, tenders, etc.

The mechanical reduction gear, while at first considered unreliable by some ship builders, has actually increased the reliability of the turbine drive, due to the smaller and more rigid construction of the turbines made possible by the use of higher speeds, rendering them less liable to distortion. Proper attention, however, must be given to the alignment of the gears and pinions, and an adequate supply of clean oil for cooling and lubrication is essential if the maximum efficiency and reliability are to be obtained.

Speed reducing gears have placed the geared turbine in a plane far above the direct connected turbine, considered from an economical standpoint, due to a saving in weight and the more favorable speed at which both propeller and turbine can be operated. There is also a distinct saving in initial and repair costs due to the great decrease in the number of turbine blades required.

No installation of hydraulic transmission has been made in the U. S. Navy, and the only reduction accomplished by electrical means is installed on the destroyer *Caldwell*, this being between the cruising and main turbines, so that very little comparative data were obtained directly. However, it is generally recognized that the efficiency of hydraulic transmission is below 90 per cent and of electrical transmission between 90 per cent and 92 per cent. Mechanical reduction efficiencies better than 97 per cent for double reduction and 98½ per cent for single reduction are now common, and this superior efficiency, combined with its simplicity, was probably the greatest factor in its development and popularity. The principal objection to the early adoption of the geared turbine was probably in the initial cost, but this is greatly outweighed by the gain in efficiency over the direct connected type.

Mechanical reduction gears are now employed having reductions of 28 to 1 for single reduction and as high as 64 to 1 for double reduction. These figures, however, do not refer to Naval vessels, the highest single reduction in the Navy being 18 to 1 and the highest double reduction being slightly above 38 to 1. The transmission of 11,250 horsepower through a single pinion and 22,500 horsepower through a single gear represents the present limit in the U. S. Navy, although powers of 18,500 horsepower and 36,000 horsepower are now being successfully transmitted through one pinion and one gear respectively by foreign Naval vessels. It is estimated that such geared installations produce an improvement in the steam consumption over direct connected turbines of the same power of from 10 per cent to 30 per cent, corresponding to full power and cruising speeds.

All reduction gears for transmitting large powers under varying conditions of torque, which are at present in use for ship propulsion, are of the double helical type using involute contour teeth. A helical gear is the only type which has proven suitable for the

transmission of large powers at high speeds. This type of gear is used to eliminate the shock common to spur gears, the load being transferred from tooth to tooth without vibration, equalizing the driving pressure over the active driving face and limiting the strain imposed at any one point. The gears are divided into two sections, having right and left hand helices inclined at equal angles to eliminate or balance the axial or end thrust.

The most suitable shape of tooth is the involute. This form of tooth can be generated automatically by a straight-sided cutting tool and gives a true line of contact on each tooth as long as the tooth is in mesh and irrespective of variations in the center distances between gear and pinion shafts due to the necessary oil clearances in bearings, wear of bearings, and adjustments which are made to provide the proper mesh or clearance between the teeth so that the oil film will not be squeezed out. This is not true with any other tooth contour. If epicycloidal teeth were used, the action would not run true if the axes of the gear and pinion remained parallel, but the center lines spread as little as .001 inch. This small variation would be detrimental to the operation due to the manner in which openings of contact occur. This form of contour, therefore, is inferior to the involute on account of the extreme sensitiveness to variations.

The helical angle of most of the earlier gears was generally 20 degrees, but due to excessive noise this angle was later increased to 45 degrees. The British Admiralty have now adopted a helical angle of 30 degrees, due largely to improvements in gear cutting machinery. There appears to be no accepted standard for this angle among American gear manufacturers, recent installations on Naval vessels ranging all the way from 23 degrees to 45 degrees, the average being about 35 degrees. The smaller angle has the advantage in efficiency and tooth strength, and, although it decreases the number of teeth in contact, it increases the length of contact with corresponding decrease in the load on the teeth. The angle of obliquity or pressure angle of the majority of gears on Naval vessels at home and abroad is 14½ degrees. No special advantage is claimed for the general use of an angle involving ½ degree, and it appears that this angle was selected to simplify computations as the sin of 14½ degrees = .250.

TYPES OF GEARS—SINGLE REDUCTION.

While some gears in Naval vessels for transmitting relatively small powers up to 2500 horsepower are of the internal planetary type, by far the greater majority are of the single reduction, one plane type, consisting of a main or slow speed gear wheel driven by one or two pinions, each pinion being connected to its respective turbine by a flexible coupling which eliminates a certain amount of misalignment. It may be pertinent to add, however, that this flexible coupling between the pinion and turbine shaft is for the purpose of allowing the pinion to take its own alignment without undue strain, and should be considered by the operating engineer as an extra precaution and not as a necessity. The alignment of the gears in a vessel should be correct without the coupling.

Gears may be classified under two general types, viz., single reduction and double reduction. Each of these classes may be subdivided into one plane or two plane type. A typical one plane single reduction gear arrangement is shown by Figure 1.

A one plane gear may be described as a gear having the driving pinions meshing on the sides of the main gear wheel and having the center lines of the gear wheel shaft, pinions and turbine rotor shafts all in the same horizontal plane. This type of gear has been very successful in Naval vessels, due probably to its simplicity, and is used for driving all types of vessels from the fast destroyers and cruisers to the slow moving auxiliary vessels.

The gear arrangement as shown by Figure 1 is used in the Navy for all Parsons, Falk, Westinghouse and De Laval geared destroyers. It is also used for the main drive of light cruisers 4 to 13, the cruising drive for battleships *Pennsylvania* and *Mississippi*, and for auxiliary vessels *Bushnell, Melville, Tippecanoe, Trinity, Dobbin, Whitney, Holland* and *Medusa*.

The Falk-driven destroyers and light cruisers have a slightly different turbine arrangement, there being an intermediate pressure turbine between the high pressure turbine and pinion.

A one plane single reduction gear driven by one turbine is also used for the cruising drive of light cruisers 4 to 6 and 9 to 13, and the battleship *Arizona*.

A two plane single reduction arrangement is shown by Figure 2. This illustration shows the center lines of the turbine rotor and

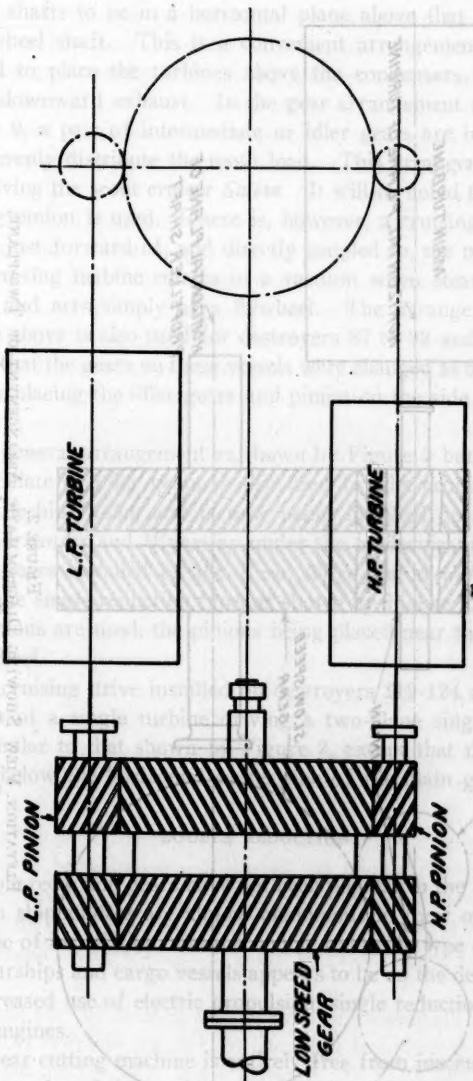


FIGURE 1.
PLAN VIEW SHOWING DIAGRAMMATIC ARRANGEMENT OF A ONE-PLANE, SINGLE REDUCTION GEAR.

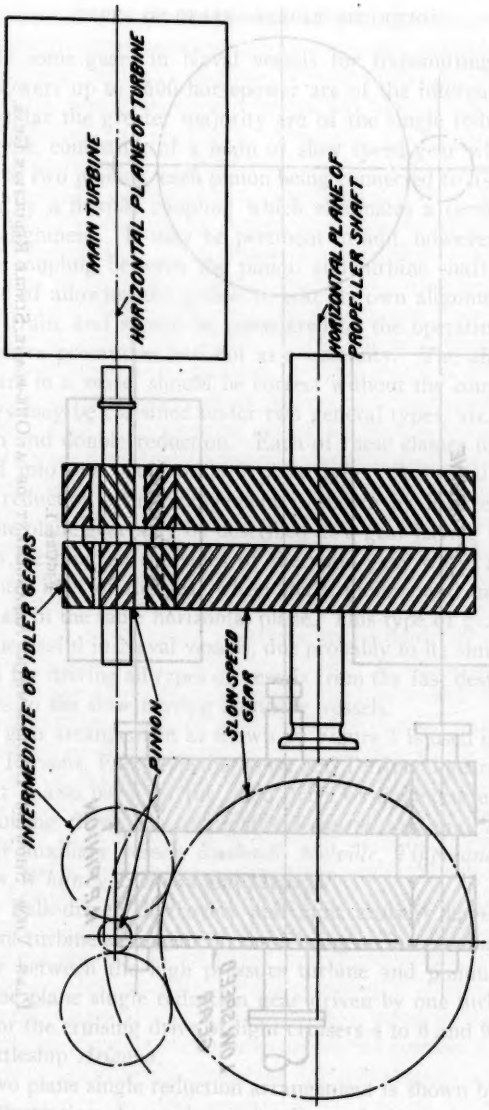


FIGURE 2.
ELEVATION VIEW SHOWING DIAGRAMMATIC ARRANGEMENT OF TWO-PLANE, SINGLE REDUCTION GEAR WITH INTERMEDIATE IDLERS.

pinion shafts to be in a horizontal plane above that of the main gear wheel shaft. This is a convenient arrangement when it is desired to place the turbines above the condensers, providing a direct downward exhaust. In the gear arrangement as shown by Figure 2, a pair of intermediate or idler gears are introduced to more evenly distribute the tooth load. This arrangement is used for driving the scout cruiser *Salem*. It will be noted that only one driving pinion is used. There is, however, a cruising turbine installed just forward of, and directly coupled to, the main turbine. The cruising turbine rotates in a vacuum when steaming at full speed, and acts simply as a flywheel. The arrangement as described above is also used for destroyers 87 to 92 and 296 to 335, except that the gears on these vessels were changed to the one plane type by placing the idler gears and pinion on the side of the main gear.

The general arrangement as shown by Figure 2 but without the intermediate or idler gears is also used for the cruising drive of the battleship *Idaho*, and is now being installed on the *Florida*, *Utah*, *Arkansas* and *Wyoming* under the modernization program.

The gears installed on the *Pyro*, *Nitro* and *Relief* are of the two-plane single reduction type, as shown by Figure 2, except that two pinions are used, the pinions being placed near the top of the gear wheel.

The cruising drive installed on destroyers 119-124 and 181-185 consists of a single turbine driving a two-plane single reduction gear similar to that shown by Figure 2, except that the pinion is placed below the horizontal center line of the main gear.

DOUBLE REDUCTION.

Double reduction gears have not been popular in the U. S. Navy. Foreign shipbuilders are almost unanimous in their opposition to this type of installation, and the double reduction type of drive for both warships and cargo vessels appears to be on the decrease, with the increased use of electric propulsion, single reduction gears and Diesel engines.

No gear cutting machine is entirely free from inaccuracies, and, as inaccuracies of the machine are reproduced in the gears, a single

reduction gear having two pinions driving directly on the main gear is bound to result in smoother operation than will be obtained with a double reduction gear having four pinions and two gear wheels. The wear on the teeth will also increase in proportion to the number of wheels and pinions used, consequently more casualties will occur in installations having double reduction gears than in single reduction units.

Until the past few years the only installation of double reduction gears in the Navy was for cruising speeds of the battleship *Nevada*. Subsequently the gunboats *Asheville* and *Tulsa* were constructed, having double reduction gears of Parsons design, as shown by Figure 3. This arrangement is known as the nested type of gear, the first and second reductions being housed in the same casing. It may be pertinent to add here that this type of arrangement has been abandoned by the Navy Department for future installations. Future double reduction gears for Naval vessels must have each reduction housed in a separate casing.

Other than the gears referred to above, the extent of double reduction installations in the Navy consists of fifteen auxiliary vessels, all of which were constructed for the U. S. Shipping Board for service in the late war and were later taken over by the Navy. All of these vessels are of the single screw type, three being of 6000 horsepower developed by a high pressure and low pressure turbine. Ten of the vessels are driven by a single turbine developing between 2500 horsepower and 3000 horsepower, and the remaining two have a single screw driven by a high pressure and a low pressure turbine developing approximately 2800 horsepower per vessel.

An illustration of the machinery arrangement of the 6000 horsepower vessels is shown by Figure 4, consisting of a one-plane gear having two intermediate or idler gears between the second reduction or low-speed pinion and main gear wheel. This arrangement shows the machinery of the U. S. S. *Argonne, Chaumont* and *Wright*.

Figure 5 illustrates the arrangement as used for the single turbine driven vessels and is typical of, but not identical to, the machinery of the U. S. S. *Antares, Vega, Spica, Denebola, Rigel, Altair, Procyon, Sirius, Capella* and *Boreas*. In some of these

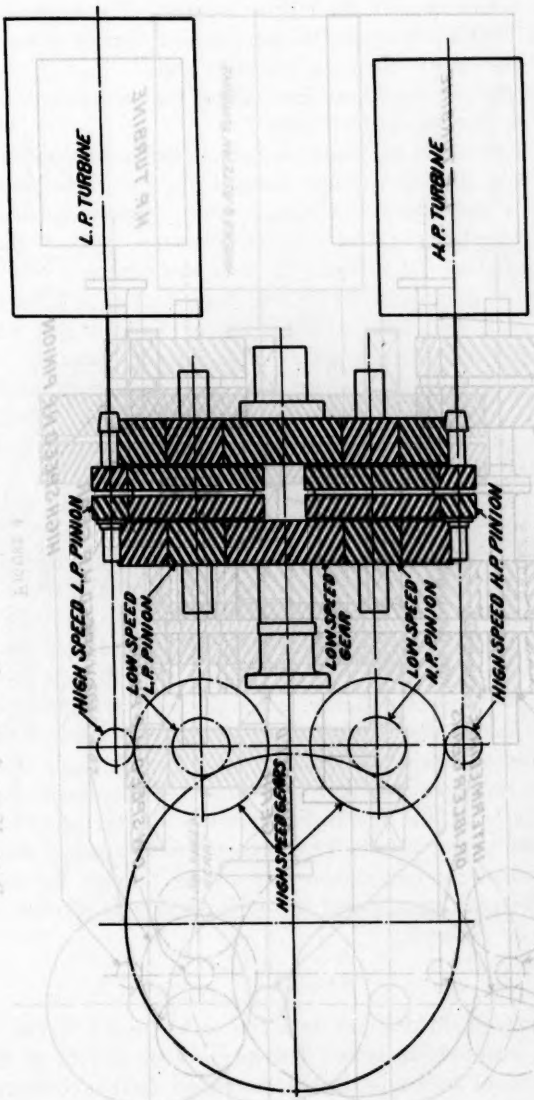


FIGURE 3.

PLAN VIEW SHOWING DIAGRAMMATIC ARRANGEMENT OF A TWO-PLANE, DOUBLE REDUCTION GEAR.

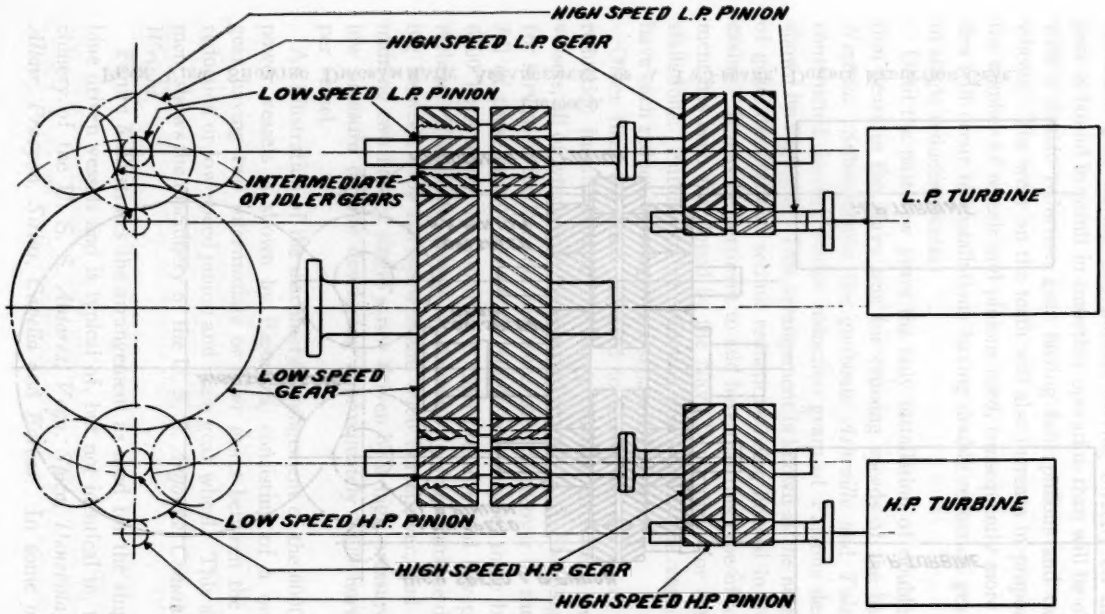


FIGURE 4.

PLAN VIEW SHOWING DIAGRAMMATIC ARRANGEMENT OF A ONE-PLANE, DOUBLE REDUCTION GEAR HAVING INTERMEDIATE OR IDLER GEARS.

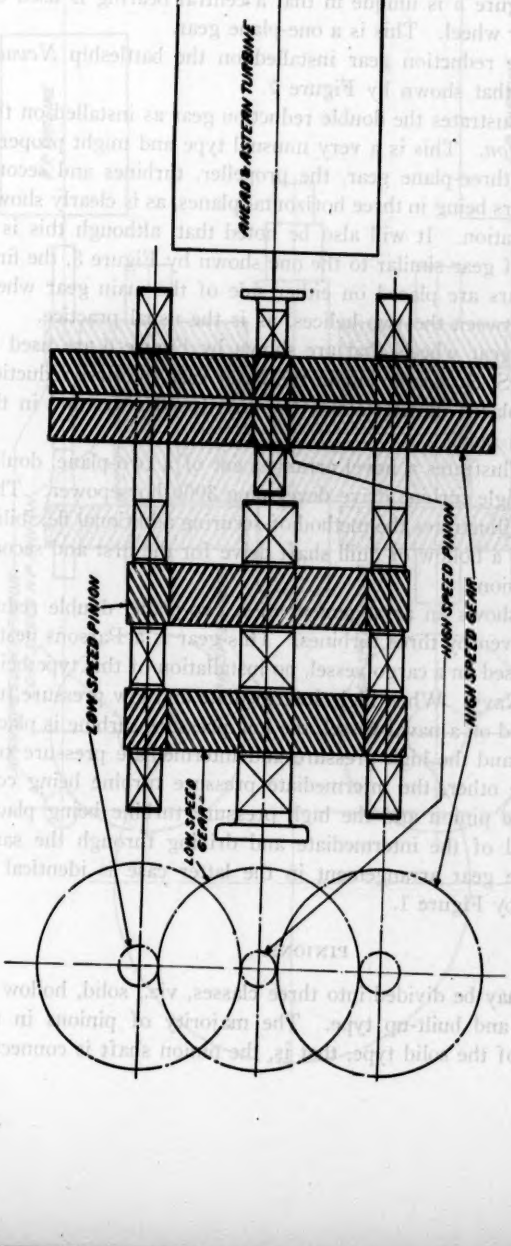


FIGURE 5.

PLAN VIEW SHOWING DIAGRAMMATIC ARRANGEMENT OF A SINGLE TURBINE, ONE-PLANE DOUBLE REDUCTION GEAR HAVING A CENTRAL BEARING FOR THE MAIN GEAR.

vessels a hollow or quill shaft drive is used. The arrangement shown by Figure 5 is unique in that a central bearing is used on the main gear wheel. This is a one-plane gear.

The double reduction gear installed on the battleship *Nevada* is similar to that shown by Figure 7.

Figure 6 illustrates the double reduction gear as installed on the U. S. S. *Yukon*. This is a very unusual type and might properly be termed a three-plane gear, the propeller, turbines and second reduction gears being in three horizontal planes, as is clearly shown by the illustration. It will also be noted that, although this is a nested type of gear similar to the one shown by Figure 3, the first reduction gears are placed on either side of the main gear wheel instead of between the two helices, as is the usual practice.

The same gear wheels that are shown by Figure 6 are used to drive the U. S. S. *Arctic*, but in the latter case the first reduction pinions are placed directly above the high-speed gears and in the same vertical plane.

Figure 7 illustrates a novel arrangement of a two-plane, double reduction, single turbine drive developing 3000 horsepower. This arrangement illustrates the method of securing additional flexibility by the use of a hollow or quill shaft drive for the first and second reduction pinions.

Figure 8 shows an arrangement of a two-plane, double reduction gear driven by three turbines. This gear is a Parsons nested type and is used on a cargo vessel, no installation of this type being used in the Navy. When high, intermediate and low pressure turbines are used on a naval vessel, the low pressure turbine is placed on one side and the high pressure and intermediate pressure turbines on the other, the intermediate pressure turbine being connected to the pinion and the high pressure turbine being placed just forward of the intermediate and driving through the same pinion. The gear arrangement in the latter case is identical to that shown by Figure 1.

PINIONS.

Pinions may be divided into three classes, viz., solid, hollow or quill drive, and built-up type. The majority of pinions in the service are of the solid type, that is, the pinion shaft is connected

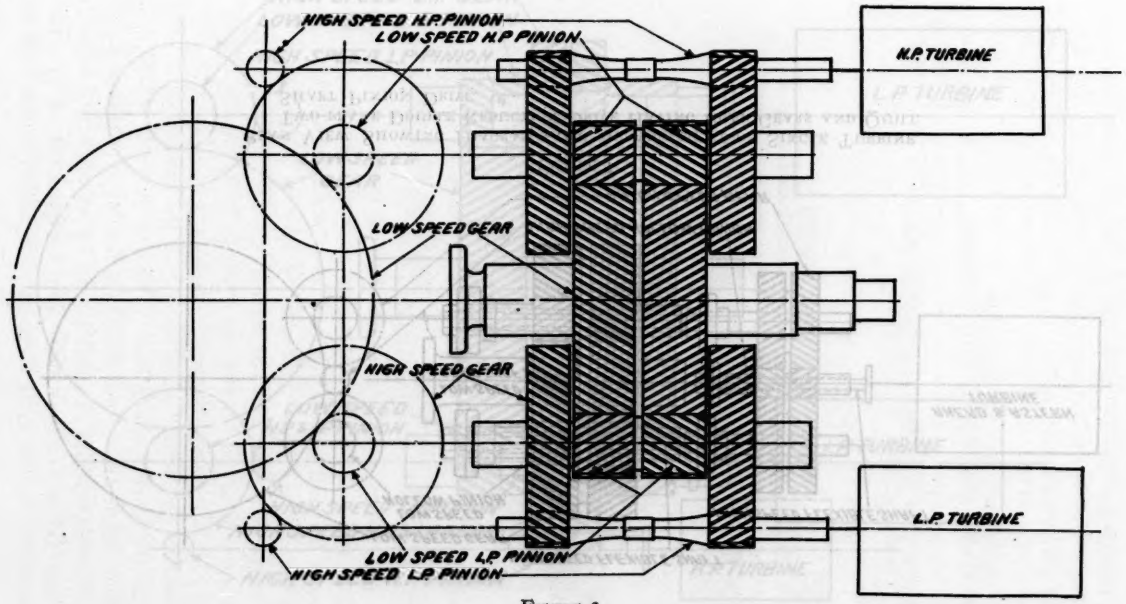


FIGURE 6.

PLAN VIEW SHOWING DIAGRAMMATIC ARRANGEMENT OF A THREE-PLANE DOUBLE REDUCTION GEAR.

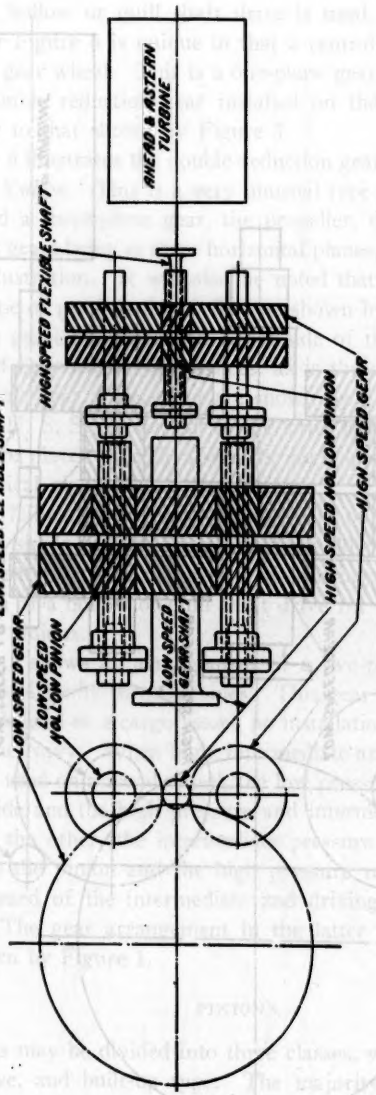


FIGURE 7.

PLAN VIEW SHOWING DIAGRAMMATIC ARRANGEMENT OF SINGLE TURBINE, TWO-PLANE DOUBLE REDUCTION DRIVE HAVING IDLER GEARS AND QUILL SHAFT PINION DRIVE.

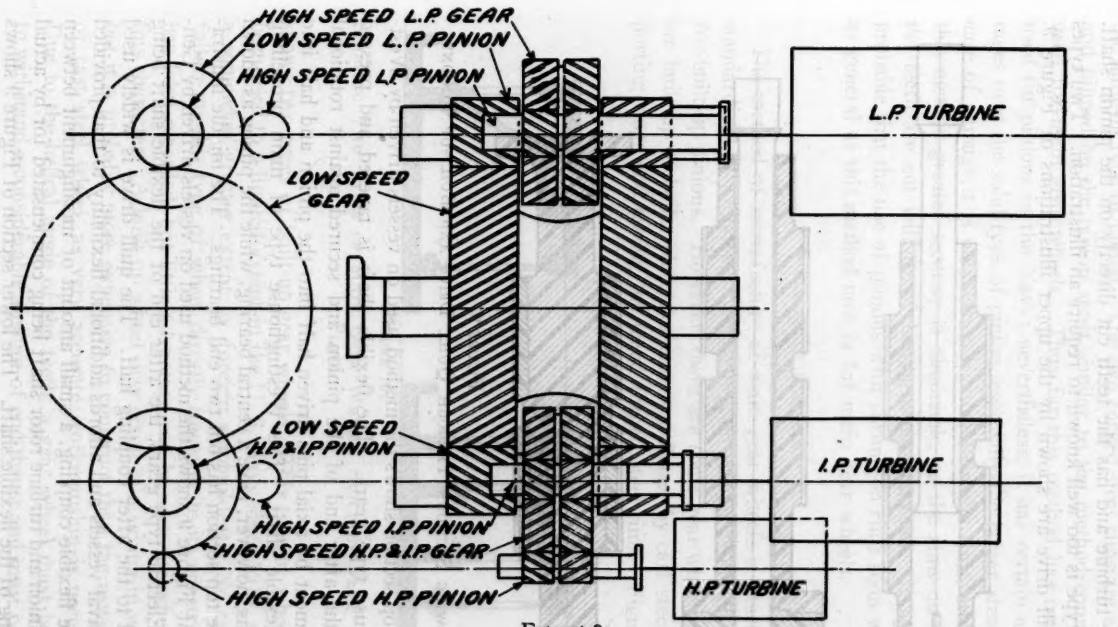


FIGURE 8.

PLAN VIEW SHOWING DIAGRAMMATIC ARRANGEMENT OF A TWO-PLANE, DOUBLE REDUCTION GEAR DRIVEN BY THREE TURBINES.

to the turbine and has the teeth cut directly on the pinion shaft. This type is too well known to require an illustration. Two types of quill drive are shown by the upper illustrations of Figure 9.

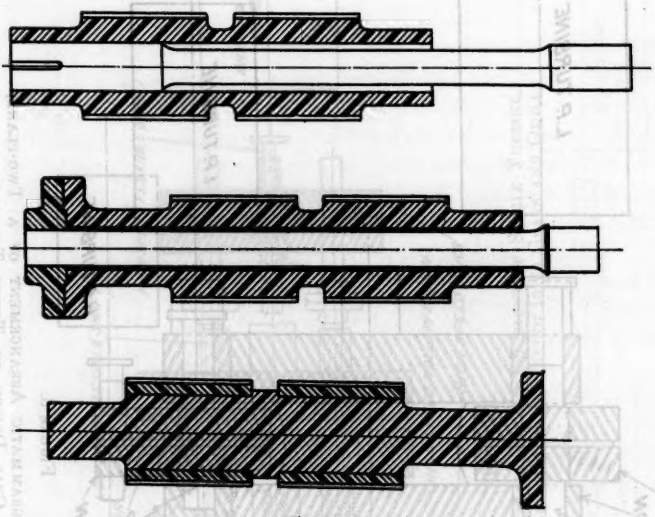


FIGURE 9.
SHOWING SECTIONS THROUGH QUILL DRIVE AND BUILT-UP PINIONS.

The top section shows the method used on vessels driven by Westinghouse type gears. The flexible shaft is tapered and pressed into the after end of the pinion and secured against rotational movement by axial pins driven half into the pinion and half into the flexible shaft. The Westinghouse type of main propelling pinions, however, have a central bearing, while the pinion as shown by the top section has but two end bearings. The middle illustration of Figure 9 shows the method used on vessels driven by General Electric type gears, the after end of the flexible shaft being keyed to the after coupling half. The quill drive is widely used on Naval vessels and affords additional flexibility to that provided by the flexible coupling, a small amount of misalignment between the pinion and turbine rotor shaft being compensated for by actual bending of the flexible shaft. The lower section of Figure 9 shows

the built-up type of pinion. The two forged steel rims are shrunk and pinned on the pinion shaft similar to the method commonly used for main gear wheels. Thus far this method has not been used for pinions in the Navy; nevertheless, it has certain advantages over the solid type of pinion, especially the larger sizes. In case of damage it is only necessary to renew the rims, and with this method greater security is obtainable, as the rims on which the teeth are cut will be of homogeneous material. It appears probable that the use of pinions with shrunk-on rims soon will be as general as this method now is for main gear wheels.

MAIN GEAR WHEELS.

The method of manufacture of main gear wheels is practically standardized, it being that of a shrunk-on rim as described above for built-up pinions. However, there are some gear wheels which are built up of boiler plate, sometimes stiffened by cast steel supporting rings and having the usual forged steel rims. Figure 10

FIGURE 10.

SHOWING TWO TYPES OF MAIN GEAR WHEELS BUILT UP FROM BOILER PLATE.

shows two types of built-up gear wheels. This type of wheel has been used to a limited extent in the U. S. Navy since 1915, and to a somewhat greater extent by foreign shipbuilders. No case of failure of this type of wheel directly traceable to its construction has been recorded, but little advantage can be seen for wheels of this character. A built-up wheel may result in a saving in weight, but it is probably more expensive to construct than the usual type of cast steel wheel, as very accurate workmanship is necessary. Due to the helical angle of the teeth, a gear wheel must be able to withstand a considerable amount of end thrust and must possess

sufficient rigidity to prevent displacement. Construction of this type is acceptable for U. S. Naval vessels, but subsequent to the pre-war destroyers no such installations have been made, and it does not appear probable that the built-up type of gear wheel will ever replace the present shrunk-on rim type.

The intermediate or idler gears, such as are shown by Figures 2 and 4, which have been used in the Navy, are of the built-up or laminated type. These gears are used only on vessels driven by the General Electric type turbines. This type of gear, manufactured under the Alquist patents, is built up of a number of steel plates or discs, sometimes mounted directly on a gear shaft and in other cases mounted on a hub, the plates being held firmly together by through studs. At their peripheries the plates are separated from one another to allow individual deflection and to equally distribute the load over the driving face of the gear. This type of gear has been installed on sixty-two vessels of the Navy in the past ten years, with but two or three failures. This construction has also been used to a limited extent for the main gear wheels. The Navy Department has, however, decided against the continued use of this type of gear, and the present laminated gears are being replaced by the "shrunk-on rim" type as fast as replacements become necessary. The main reason for this change is that simplicity in construction provides greater reliability in operation. The construction of the laminated type of gear is illustrated by Figure 11.

In reduction gears, as in any mechanical device, simplicity reflects directly in increased reliability. Previous designs incorporating such components as laminated gears, intermediate or idler

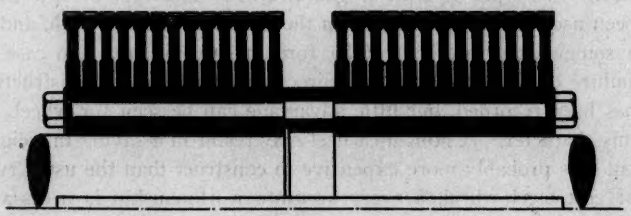


FIGURE 11.
SHOWING CONSTRUCTION OF THE LAMINATED TYPE OF GEAR.

gears, quill drive pinions and flexible pinion supports, may have served their purpose in providing increased flexibility and more even distribution of tooth load, but it appears that the advantage in flexibility gained by the departure from the simplest form of successful mechanical reduction is more than balanced by the decrease in reliability, without considering the increased cost of production.

With the advent of greater accuracy in gear cutting machines and the adoption of more suitable materials, the trend is again in the direction of simplicity. The laminated gear, or the gear built up from a series of plates, is already a thing of the past, and the future installation of gears having intermediates or idlers is very remote. They are definitely prohibited for future naval gears. The quill or flexible shaft driven pinion, and the pinion having flexible supports, although still popular with some gear manufacturers, is bound to be replaced in the near future by the pinion directly connected to the turbine by a suitable flexible coupling. Serious consideration has already been given to the elimination of the quill drive from the gear designs which are acceptable to the Navy Department for installation on U. S. Naval vessels.

It is always desirable to fit a middle bearing to a pinion shaft although it is not always a necessity. Pinions on a great many destroyers now in service have two central bearings, or a total of four bearings to a pinion. The elimination of the middle bearing on a small diameter pinion having a relatively long tooth face may cause excessive bending deflection. The Navy Department has specified that for pinions without a central bearing such as are used with generating or auxiliary turbines, the total length of tooth face plus the distance between the helices shall not exceed 2-1/2 times the pitch diameter of the pinion. A middle pinion bearing is required for main propelling gears of U. S. Naval vessels.

Gears should be so designed that the number of teeth in the main gear wheel is not a multiple of the number of teeth in the pinion, or in other words, if the ratio between the gear and pinion is not a whole number, the same teeth will not come in contact on successive revolutions, thus insuring uniform wear on both.

It has long been the practice of the majority of gear manufacturers to chamfer the ends of pinion teeth and to thin the ends of gear teeth for a short distance to prevent the breaking off of unsupported edges. This has previously been required for U. S. Naval gears. The Navy Department now specifies that thinning and chamfering of gear and pinion teeth will be permitted.

The practice of thinning the gear teeth is rather an expensive process, especially on large gear wheels, and the benefits therefrom are questionable. The Committee of the American Gear Manufacturers Association has agreed that it is better practice to leave the teeth untouched, and that when teeth are thinned the unusual wear between the gear and pinion has a tendency to break the corners of the teeth due to the ridge which will form. They recommend that teeth of both gear and pinion be turned square, not beveled, except for a slight radius on the outer corners to obviate bruising of blanks when they are being handled. The square ends are preferred over the beveled as they tend to wear evenly and avoid ridges such as are caused by overlapping edges.

MATERIALS.

There is a great difference of opinion among the gear manufacturers as to the most suitable materials for reduction gears, especially pinions. For gear casings the practice of using steel castings which have been thoroughly annealed is almost universal, although for future construction gear casings of a cast aluminum alloy will be accepted for U. S. Naval gears. This is an alloy having an aluminum content of 84 per cent to 94-1/2 per cent and a Brinell hardness of from 45 to 80. Boiler plate is also acceptable for built-up casings.

Gear wheel centers are usually of cast steel. High test gray iron or semi-steel having a minimum tensile strength of 30,000 pounds is also considered suitable.

The material for gear wheel rims which are to be shrunk on is generally a low carbon steel having a tensile strength of from 60,000 pounds to 85,000 pounds. The rims are usually forged, but there appears to be an increasing sentiment in favor of rolled rims in place of forgings.

A great variety of steels are used for pinions, including chrome nickel and chrome vanadium steels having a tensile strength of from 80,000 pounds to 145,000 pounds. The majority of manufacturers, however, use a 3-1/2 per cent nickel steel having a tensile strength of 100,000 pounds and a Brinell hardness of 180 to 200. The minimum requirement for pinion material for U. S. Naval gears is a .45 carbon steel forging having a tensile strength of 80,000 pounds, a yield point of 50,000 pounds, and an elongation of 25 per cent in 2 inches. The general trend is towards softer materials for pinions, and the adoption of a low carbon steel for pinions, similar to that now generally used for gear rims, is predicted.

CONSTRUCTION.

There are some phases in the design and construction of reduction gears that are considered essential if the gear is to give satisfactory operation over a long period of service, and which all gear manufacturers have not recognized as necessary or desirable to the extent of their being adopted as a standard practice or method. This may be due, in some cases, to the fact that the gears, after installation, operated satisfactorily throughout the guarantee period, such deficiencies of a minor character as were apparent after this period being remedied by the engineer force of the vessel, or at a yard, without being brought to the attention of the gear manufacturers. Some of the more important items which are sometimes overlooked in the design and construction of gears, and without which the maximum efficiency and freedom from breakdowns over long periods of continual operation will seldom be obtained, will be briefly described.

Where flexible couplings between the pinion and turbine shaft are fitted inside the gear casing, they should be enclosed by a coupling guard to prevent oil being thrown against the gear casing. Without such a guard the flying oil will, in time, cause considerable wear on the casing if the coupling rotates at high speed. The coupling guard, if fitted, should be split to permit ready removal and should be provided with holes for draining the oil collected to the gear case.

Gear casings should be so strongly ribbed that they will absorb all stresses developed by the gear under the maximum conditions of operation, and extreme care should be taken in boring and fitting bearings. This is especially desirable in the fixed bearing type, as the accuracy of boring and fitting bearings governs the degree of accuracy of the alignment.

A necessary provision, which is often omitted from a gear design, is that of draining the main gear wheel. A gear wheel must have a sufficient number of ribs or webs to provide the necessary rigidity. These webs, in practically all cases, have lightening holes cored in the sides. It is customary to fit covers over these holes to exclude lubricating oil. Even with these covers in place a small amount of oil is always liable to enter the interior of the gear wheel and accumulate until sufficient in quantity to seriously affect the balance of the gear, especially if the gear operates at a speed such as is used for a destroyer or a light cruiser. To prevent this accumulation of oil, radial holes should be drilled through the gear wheel rim in sufficient number and in such location that the oil can be effectively drained from the interior of the wheel.

Gears should be balanced both statically and dynamically before installation, especially the high speed elements. It has been found that noise in such small gears as are used in lathes disappears and that less power is required for their operation after the gears have been removed and balanced.

Oil spray nozzles for astern operation are required for U. S. naval vessels, probably, more in the nature of an extra precaution than a necessity. British vessels are not required to be fitted with astern spray nozzles, as the periods of astern operation are short and there will always be sufficient spray or vapor present in the casing to furnish lubrication for astern operation. In fact, gears have been successfully operated having oil vapor lubrication in place of spray nozzles. In the design of spray nozzles, however, it is well to remember that a few large spray holes are preferable to a larger number of small holes, as the oil passage is less liable to become obstructed.

Gears should be provided with a hand turning device for use in examination. This is also very essential for producing the torque necessary to secure proper alignment and for jacking over the

turbines. The turning device should be so designed that it may be quickly assembled, and disassembled for stowage when not in use.

The main gear shaft should be provided with a clearance indicator to detect any axial movement of the shaft caused by wear of the main thrust shoes. Figure 12 shows a satisfactory type of clearance indicator, the construction of which is self explanatory.

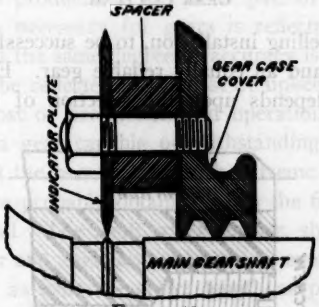


FIGURE 12.

CLEARANCE INDICATOR FOR DETECTING AXIAL MOVEMENT OF GEAR SHAFT.

Gear wheel rims which are shrunk on the gear hubs should be securely doweled to prevent their displacement. This is usually accomplished by a suitable number of axial tapped holes, half of the hole being in the inside of the gear rim and half in the periphery of the gear center. Plugs are then screwed into the holes so that their heads are slightly below the surface, the metal of the wheel then being peened over the ends of the plugs to prevent their backing out. Figure 13 shows a satisfactory method of accomplishing this.

It is frequently necessary that studs or pins be tightly screwed or driven into holes where the metal beneath the hole is thin, and care must be taken that the stud or pin does not break through, due to the presence of an oil reservoir, or a steam space in the case of a turbine. In numerous cases of this kind attempts to tighten a stud or drive home a tight-fitting pin has resulted in the bottom of the hole being forced out into the oil reservoir or steam space. It may be readily seen that as a stud is being tightened or a pin is nearing the bottom of the hole, the air in the hole has no way

to escape, and it becomes highly compressed, either preventing the stud or pin from reaching the bottom of the hole or blowing out the thin metal. A simple prevention for this is to file a groove along the threads of the stud with a three-cornered file or to drill a 1/32 inch hole through the center of the pin to prevent the compression of air.

GEAR CUTTING.

A geared propelling installation, to be successful, must have an efficient turbine and an equally reliable gear. Efficient and quiet gear operation depends upon the selection of proper materials,

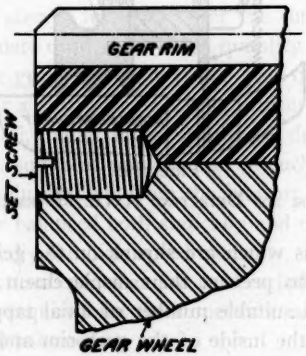


FIGURE 13.

SHOWING METHOD OF SECURING GEAR RIMS AGAINST DISPLACEMENT.

correct design and a high degree of accuracy in cutting to produce the designed helical angle, tooth contour and pitch. The gears must be properly aligned and lubricated. It has been the practice to select the number of teeth and helical angle that will bring the maximum number of teeth in contact. The greater the helical angle the more teeth will be in contact at one time, but by increasing the number of teeth in contact, the length of contact, the efficiency and strength of tooth is decreased in proportion. The helical angle in general use has been reduced from 45 degrees to 35 degrees, but the future adoption of a standard helical angle of 25 degrees, or even 20 degrees, appears probable. Quiet and

smooth running gears require extremely accurate gear cutting apparatus for their production, and the gear cutting machines which are capable of producing the type of gears now in demand require greater accuracy than is necessary for probably any other type of machine tools. It is a recognized fact that manufacturers in the United States produce gear cutting machines of greater accuracy than those produced in any other part of the world. The extreme accuracy necessary for gears is reflected in the cost of production, and if the same degree of accuracy is to be maintained in service it will be reflected in the cost of upkeep. Future relief from the high cost of production and operation depends on the development of a gear capable of withstanding long periods of operation without the necessity of such extreme accuracy.

Best results are probably obtained when the final cut from the teeth is very small. The teeth, after cutting, should be carefully tested for contour and angle and to insure an even contact surface, which should be as near the entire length as possible. Various means are employed for securing a good contact, such as running with a dummy pinion, using an abrasive, hand scraping and filing, but it is the general opinion that best results are obtained by mechanical polishing to remove inequalities left by the cutting tool, and prohibiting any method of hand fitting.

OPERATION.

The extreme care which must be exercised in the alignment of a gear during installation is second only to the accuracy required in manufacture. Accurately cut gears which are improperly aligned will not operate in a quiet and satisfactory manner. Engineers should thoroughly understand that gears which have been carefully aligned as they rest in the bearings will not operate to advantage, as their normal operating condition has not been satisfied. Referring to a simple single reduction arrangement such as that shown by Figure 1, the pinions will be in alignment under operating conditions when one pinion rests against the lower half of the bearing shell while the other pinion is against the top half of its shell, allowance being made for the oil film. Turbine rotors, due to their weight and the fact that forces acting on the blading are equal, always rest against the lower half of the bearings, but with

two pinions driving a gear wheel the forces acting on one pinion will be in an upward direction while the forces acting on the other pinion will be downward. The main gear shaft, like the turbine rotor, will rest against the lower halves of the bearings due to its weight and the fact that the forces exerted by the pinions are nearly equalized.

It is apparent, therefore, that the pinions, to operate satisfactorily and without undue wear and noise, must be in their natural operating position during alignment. This may be accomplished by applying a torque to the pinions in the direction of ahead rotation, which can be done on most installations by means of the hand turning gear. Under no circumstances should the gear casing joints be used in securing alignment as the gear casing joint flanges are often distorted by the bolts being set up.

A simple gauge for checking the alignment of the turbine rotor shaft and pinion shaft is shown by Figure 14.

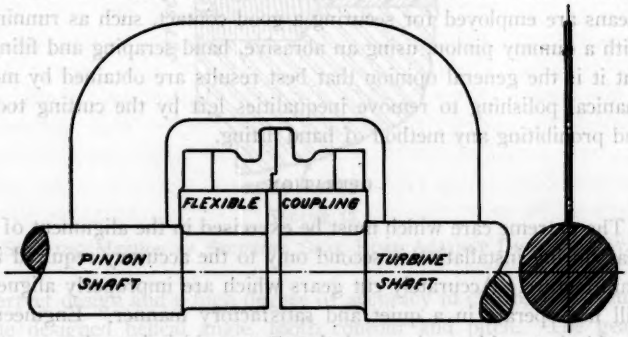


FIGURE 14.
ALIGNMENT GAUGE.

This gauge is inexpensive to manufacture and its use is very simple. A small test shaft should be included for the purpose of checking the accuracy of the gauge. When not in use the test shaft and gauge should be carefully stowed away to prevent damage.

Gears, due to the involute tooth contour, may be operated without any disadvantage when the center-to-center distance of the

gear and pinion is somewhat more than specified by the design, but this fact should not lead an operator to believe that, after the bearings have been renewed or rebabbitted, the gears may be meshed closer than called for by the designs, for by so doing the oil film may be squeezed out from between the teeth, causing serious damage.

When pinions are to be removed from their bearings, care should be taken to note the marking on the ends of the teeth in order that the gears may be reassembled in the same position. The ends of gear and pinion teeth are usually marked, but if no assembly marks are found, the teeth should be marked in a similar manner to that shown by Figure 15. Care should also be taken to mark

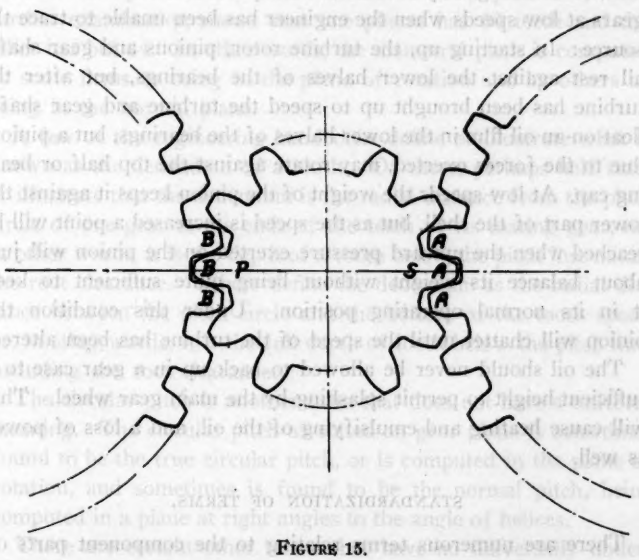


FIGURE 15.

METHOD OF MARKING THE ENDS OF GEAR TEETH TO FACILITATE ASSEMBLING AFTER REPAIRS.

the ends of the teeth on new pinions which are fitted. The alignment of the flexible coupling, pinions and gears should be checked each time the gear is reassembled after repairs.

Noise and pounding which may develop in a gear after considerable operation may usually be traced to excessive clearances or misalignment, and should be eliminated at the first opportunity, to save wear on the bearings and flexible coupling. A noise of this character may be easily located by an experienced operator, but there are times when an unusual pounding may develop at certain speeds, the source of which can not be traced, and is sometimes assumed to be due to misalignment or the presence of foreign particles in the gear. Examination after the gear case has been opened will show that no chips or foreign particles have entered the case, the bearings have not been wiped and the gear is found to be properly aligned.

The following may serve as an explanation of cases of noisy gears at low speeds when the engineer has been unable to trace the source: In starting up, the turbine rotor, pinions and gear shafts all rest against the lower halves of the bearings, but after the turbine has been brought up to speed the turbine and gear shafts float on an oil film in the lower halves of the bearings, but a pinion, due to the forces exerted, may rotate against the top half or bearing cap. At low speeds the weight of the pinion keeps it against the lower part of the shell, but as the speed is increased a point will be reached when the upward pressure exerted on the pinion will just about balance its weight without being quite sufficient to keep it in its normal operating position. Under this condition the pinion will chatter until the speed of the turbine has been altered.

The oil should never be allowed to back up in a gear case to a sufficient height to permit splashing by the main gear wheel. This will cause heating and emulsifying of the oil, and a loss of power as well.

STANDARDIZATION OF TERMS.

There are numerous terms relating to the component parts of gear teeth which are used by all gear manufacturers but which do not have the same meaning to all of them, and it would be advantageous if such terms and definitions were reduced to a common standard. The present nomenclature as used by the various manufacturers may be the cause of some confusion and loss of time in the examination and comparison of designs which may be sub-

mitted by various companies desiring to furnish propelling machinery for a certain vessel. It would simplify and expedite the shipbuilder's examination of designs if the nomenclature and computations as listed by the plans had a definite meaning regardless of the manufacturer from whom the data were obtained. It would also facilitate in the comparison of various gears by a company operating a fleet of vessels having gears furnished by various firms.

Examination shows that recognized engineering handbooks do not agree upon the meaning and computation of various and universally used terms; for example, the term "diametral pitch" is in all cases understood to be a ratio of the pitch diameter to the number of teeth, but two plans from different firms referring to the same size pinions may state that the diametral pitch is 5, while further examination shows that one pinion has 50 teeth and the other has but 44. In other words, the diametral pitch is computed by some firms as being in the plane of rotation and by others as being in the normal plane.

A few of the engineering handbooks define or illustrate what is known as the dedendum of a tooth. Of these, perhaps half state or illustrate that the dedendum is the radial distance from the pitch circle of one gear to the ends of the teeth of its mating gear, or, in other words, it is the depth of a tooth below the pitch line minus the root clearance. Others define the dedendum as the radial distance between the pitch circle and the bottom of the tooth space, or, in other words, it is the full depth of tooth below the pitch line, including the root clearance.

The circular pitch is another term that does not have a uniform meaning. The circular pitch as stated on gear plans is sometimes found to be the true circular pitch, or is computed in the plane of rotation, and sometimes is found to be the normal pitch, being computed in a plane at right angles to the angle of helices.

There are certain other terms that have no universally understood meaning, and, in view of this fact, the following definitions of the more commonly used gear terms relating to helical gears and the method of computing them have been prepared by the author and have been accepted as standard nomenclature by the Bureau of Engineering of the Navy as a common standard which

must be used by all shipbuilders and turbine and gear manufacturers in computing the table of gear cutting data which appears on all gear plans submitted to the Navy Department.

The following definitions are illustrated by Figure 16:

$A =$ Pitch diameter $= B - 2E$	Pitch diameters are imaginary circles, the diameters of which are the same as the diameters of a pair of friction gears that would replace the toothed gears.
$B =$ Outside diameter $= A + 2E$	Outside diameter is the diameter measured over the ends of the teeth.
$C =$ Root diameter $= A - 2F$	Root diameter is the diameter of the bottom land joining adjacent teeth.
$D =$ Diametral pitch $= \frac{G}{A}$	Diametral pitch is the ratio of the number of teeth to the pitch diameter, taken in the plane of rotation.
$E =$ Addendum $= \frac{B - A}{2}$	Addendum is the radial distance from the pitch circle to the ends of the teeth.
$F =$ Dedendum $= \frac{A - C}{2}$	Dedendum is the radial distance between the pitch circle and the bottom of the tooth space.
$G =$ Number of teeth $= A \times D$	
$H =$ Angle of helices	
$J =$ Normal pressure angle	Normal pressure angle is the angle between the common tangent to the two base circles (gear and pinion) and a line perpendicular to their common center lines, taken in a plane at right angles to the angle of helices.

K = Chordal thickness	Chordal thickness is the thickness of a tooth measured on the chord of the pitch diameter, at right angles to the angle of helices.
L = Normal pitch = M cos. H	Normal pitch is the length of arc on the pitch circle between similar points of adjacent teeth, taken in a plane at right angles to the angle of helices.
M = Circular pitch $= \frac{\pi \times A}{G}$	Circular pitch is the length of arc on the pitch circle between similar points of adjacent teeth, taken in the plane of rotation.
N = Root clearance = F − E′	Root clearance is the space between the end of a tooth and the bottom of its mating space.
O = Face	Face is that part of a tooth profile which lies between the pitch circle and the end of a tooth.
P = Flank	Flank is that part of a tooth profile which lies between the pitch circle and the bottom of the tooth space.
R = Working depth = E + E′	Working depth is the depth to which the teeth of one gear enter into their mating space.
S = Module $= \frac{1}{D}$	Module is the ratio of the pitch diameter to the number of teeth. It is the reciprocal of the diametral pitch.
T = Pitch point	Pitch point is the point of tangency of the pitch circles of two mating gears.
U = Whole depth = E + F	Working depth is the total radial depth of space between the teeth.
V = Base circle	Base circle is the circle from which a line is unwound to generate the involute curve.

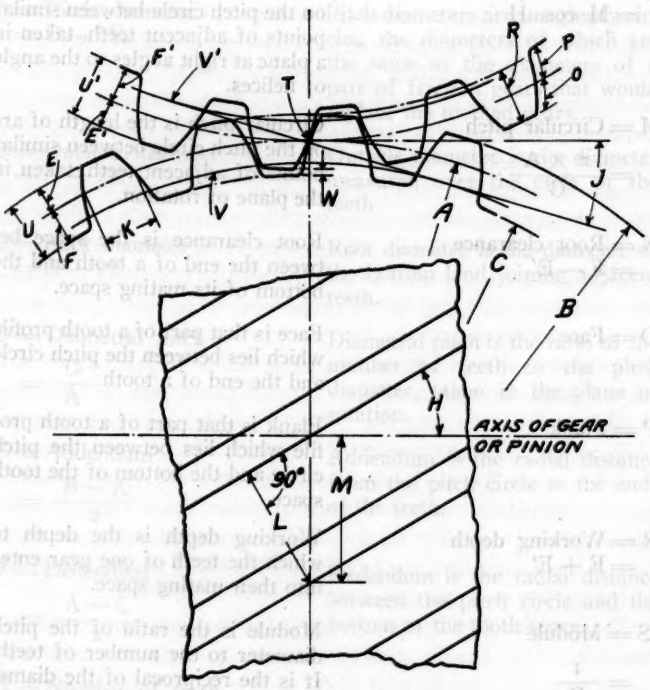


FIGURE 16.

REDUCTION GEAR NOMENCLATURE.

DESTROYERS	MAKE OF GEAR	PART	PITCH DIAMETER	OUTSIDE DIAMETER	ROOT DIAMETER	DIAMETRAL PITCH	ADDENDUM	DEDENDUM	NUMBER OF TEETH	HELICAL ANGLE	NORMAL PRESSURE ANGLE	CIRCULAR PRESSURE ANGLE	NORMAL PITCH	CIRCULAR PITCH	THICKNESS OF TOOTH ON PITCH DIA.	ROOT CLEARANCE	S.H.P. TRANSMITTED	R.P.M.	REDUCTION RATIO
75-78 131-135 342-347	PARSONS SINGLE	H.P. PINION	9.916"	10.267	9.3394	4.135	.0885	.2883	41	35°52.3'	14°30'	14°37'	.5831	.7598	.281"	.1028	6050	9036	6.85 to 1 ●
		L.P. PINION	16.688	17.059	16.1114	4.135	.0885	.2883	69	35°53.3'	14°30'	14°37'	.5831	.7598	.281"	.1028	6050	1804	
		GEAR	67.962	68.333	67.3854	4.135	.0885	.2883	281	35°53.3'	14°30'	14°37'	.5831	.7598	.281"	.1028	12100	443	
79-86, 95-112 161-180 251-295	FALK CO. SINGLE	H.P. PINION	10.800	11.200	10.200"	2.5	.240"	.300"	27	44°53.5'	18°15'	25°	.8899	1.2566	.435"	.060"	6850	9216	6.66 to 1
		L.P. PINION	11.600	12.000	11.000"	2.5	.240"	.300"	29	44°53.5'	18°15'	25°	.8899	1.2566	.435"	.060"	6600	2995	
		GEAR	77.200	77.600	76.600"	2.5	.240"	.300"	193	44°53.5'	18°15'	25°	.8899	1.2566	.435"	.060"	13450	450	
87-92	G.E. CO. SINGLE	PINION	8.833"	9.043	8.559"	6	.105"	.137"	53	27°	20°	22°13'	.9665	.5236	.228"	.032"	13500	3482	7.74 to 1
		IDLERS	26.167"	26.377	25.893"	6	.105"	.137"	157	27°	20°	22°13'	.9665	.5236	.228"	.032"	6750	1170	
		GEAR	68.333"	68.543	68.059"	6	.105"	.137"	410	27°	20°	22°13'	.9665	.5236	.228"	.032"	13500	450	
93-94 136-141 336-341	PARSONS SINGLE	H.P. PINION	9.916	10.156	9.316"	4.135	.120"	.200"	41	35°53.3'	14°30'	18°37'	.5831	.7598	.281"	.080"	6050	9025	6.83 to 1 ●
		L.P. PINION	16.688	16.928	16.288"	4.135	.120"	.200"	69	35°53.3'	14°30'	18°37'	.5831	.7598	.281"	.080"	6050	1798	
		GEAR	67.721	67.961	67.321"	4.135	.120"	.200"	280	35°53.3'	14°30'	18°37'	.5831	.7598	.281"	.080"	12100	443	
[illegible] 142-160 206-230	DE LAVAL SINGLE	H.P. PINION	9.916	10.156	9.4466	4.135	.120"	.2347	41	35°53.3'	13°	22°50'	.5831	.7598	.281"	.1147	6300	9036	6.85 to 1 ●
		L.P. PINION	16.688	16.928	16.2186	4.135	.120"	.2347	69	35°53.3'	13°	22°50'	.5831	.7598	.281"	.1147	6300	1804	
		GEAR	67.962	68.202	67.4926	4.135	.120"	.2347	281	35°53.3'	13°	22°50'	.5831	.7598	.281"	.1147	13000	443	
119-124	NEWPORT	PINION	7.459	7.870	6.922"	3.8675	.1855"	.2885"	29	44°8.2'	15°30'	21°7'	.5829	.81233	.272"	.1030	1800	3159	9.97 to 1
181-185	SINGLE	GEAR	74.327	75.098	74.150"	3.8675	.1855	.2885	288	44°8.2'	15°30'	21°7'	.5829	.81233	.249"	.1030	1800	321	
186-199 231-250	WESTINGH. SINGLE	PINIONS	10.287	10.758	9.378"	3.031	.2855	.4265	31	30°	14°30'	16°58'	.897	1.036"	.446	.141"	6750	2918	6.48 to 1
		GEAR	66.312	66.883	65.353"	3.031	.2855	.4595	201	30°	14°30'	16°58'	.897	1.036"	.423"	.141	13500	450	
296-335	G.E. CO. SINGLE	PINION	8.800	9.044	8.484"	5	.122"	.158"	44	35°	16°30'	20°	.5147	.6283	.249"	.036"	13500	3481	7.75 to 1
		IDLERS	24.200	24.444	23.884"	5	.122"	.158"	131	35°	16°30'	20°	.5147	.6283	.249"	.036"	6750	1171	
		GEAR	68.200	68.444	67.884"	5	.122"	.158"	341	35°	16°30'	20°	.5147	.6283	.249"	.036"	13500	450	

● NOTE: S.H.P. 135 & 336 TO 344 SAME AS 113-118 CLASS
S.H.P. 206 TO 230 SAME AS 186-199 CLASS

FIGURE 17.

COMPARISON OF DESTROYER GEARS.

VESSEL	MAKE OF GEAR	PART	PITCH DIAMETER	OUTSIDE DIAMETER	ROOT DIAMETER	DIAMETRICAL PITCH	ADDENDUM	DEDENDUM	NUMBER OF TEETH	HELICAL ANGLE	NORMAL PRESSURE ANGLE	CIRCULAR PRESSURE ANGLE	NORMAL PITCH	CIRCULAR PITCH	THICKNESS OF TOOTH ON PITCH DIA.	ROOT CLEARANCE	S.H.P. TRANSMITTED	R.P.M.	REDUCTION RATIO
SALEM	G.E. CO. SINGLE	PINION	7.400	7.644	7.084	5	.122"	.158"	37	35°	16°30'	20°	.5147	6283	.252"	.036	10000	3810	10.03 to 1
		IDLERS	24.200	24.444	23.884	5	.122"	.158"	131	35°	16°30'	20°	.5147	6283	.252"	.036	5000	1076	
		GEAR	74.200	74.444	73.884	5	.122"	.158"	371	35°	16°30'	20°	.5147	6283	.252"	.036	10000	380	
LIGHT CRUISERS 4-6	WESTINGH SINGLE	CRUIS. PIN	4.474	4.724	4.124	6.928	.125"	.175"	31	30°	14°30'	16°38'	.3927	.4535	.188"	.050	6400	6285	5.61 to 1
		CRUIS. GEAR	25.115	25.365	24.157	6.928	.125"	.179"	174	30°	14°30'	16°38'	.3927	.4535	.196"	.054	6400	1119	
		M. PINIONS	12.537	13.108	11.708	3.031	.2855	.4145	38	30°	14°30'	16°38'	.8976	1.0365	.446"	.129	11250	2522	6.82 to 1
		M. GEAR	85.448	86.019	84.659	3.031	.2855	.3945	259	30°	14°30'	16°38'	.8976	1.0365	.448"	.109	11250	370	
LIGHT CRS. 7 & 8	FALK CO. SINGLE	PINIONS	13.429	13.829	12.915	3.5	.200"	.257"	47	30°	21°55'	25°	.774"	.8978	.386"	.057	11250	2332	6.28 to 1
		GEAR	84.286	84.686	83.772	3.5	.200"	.257"	295	30°	21°55'	25°	.774"	.8978	.386"	.057	11250	370	
LIGHT CRUISERS 9-13	DE LAVAL SINGLE	CRUIS. PIN	6.600	6.8827	6.2723	5	M135	M6385	33	44°58'43"	14°30'	20°5'	.4444	.6283	.221"	.0225	900	9002	3.15 to 1
		CRUIS. GEAR	20.800	21.0827	20.4723	5	M135	M6385	104	44°58'43"	14°30'	20°5'	.4444	.6283	.221"	.0225	900	1270	
		H.P. PINION	12.726	12.966	12.3566	.3859035	.120"	.2347	49	44°23'419	14°30'	15°535	.583"	.815919	.281"	.1147	11250	2515	6.83 to 1
		L.P. PINION	21.566	21.796	21.0866	.3859035	.120"	.2347	83	44°23'419	14°30'	15°535	.583"	.815919	.281"	.1147	11250	1486	
		GEAR	87.005	87.245	86.5356	.3859035	.120"	.2347	335	44°23'419	14°30'	15°535	.583"	.815919	.281"	.1147	11250	368	
NEVADA	G.E. CO. DOUBLE	H.S. PINION	6.500	6.772	6.156	6	.136"	.172"	39	35°	16°30'	20°	.4289	.5236	.212"	.036	1750	3196	23.85 to 1
		H.S. GEAR	25.167	25.439	24.823	6	.136"	.172"	151	35°	16°30'	20°	.4289	.5236	.212"	.036	875	825	
		L.S. PINION	10.000	11.000	9.200	2.5	.500"	.600"	25	23°	14°30'	15°42'	1.1567	1.2566	.617"	.100"	875	825	
		L.S. GEAR	61.600	62.200	60.400	2.5	.300"	.600"	154	23°	14°30'	15°42'	1.1567	1.2566	.515"	.100"	1750	134	
PENNSYLVA	WESTINGH SINGLE	CR. PINIONS	4.691	5.145	4.439	5.182	.257"	.096"	24	30°	14°30'	16°38'	.525"	.606"	.263"	.019"	750	2258	15.09 to 1
		CR. GEAR	69.654	69.808	69.102	5.182	.077"	.276"	361	30°	14°30'	16°38'	.525"	.606"	.263"	.019"	1500	150	
MISSISSIPPI	WESTINGH SINGLE	CR. PINIONS	5.017	5.529	4.821	5.182	.256"	.098"	26	30°	14°30'	16°38'	.525"	.606"	.260"	.021"	750	2322	13.89 to 1
		CR. GEAR	69.653	69.808	69.096	5.182	.077"	.278"	361	30°	14°30'	16°38'	.525"	.606"	.261"	.022"	1500	160	
ARIZONA	WESTINGH SINGLE	CR. PINION	6.235	6.635	5.775	4.3305	.200"	.230"	27	30°	14°30'	16°38'	.6283	.7255	.314"	.030"	2000	2015	12.59 to 1
		CR. GEAR	78.520	78.920	77.970	4.3305	.200"	.275"	340	30°	14°30'	16°38'	.6283	.7255	.313"	.075"	2000	160	
IDAHO	DE LAVAL SINGLE	CR. PINION	7.5917	7.7717	7.2117	3.851	.120"	.160"	29	44°23'7	15°	17°392	.58304	.815919	.281"	.040"	2000	2.500	13.07 to 1
		CR. GEAR	98.4378	98.6178	98.1178	3.851	.120"	.160"	379	44°23'7	15°	17°392	.58304	.815919	.281"	.040"	2000	191	
FLA, UTAH ARK, WYO.	NEWPORT SINGLE	CR. PINION	7.025	7.396	6.436	4.1287	.1855	.2955	29	40°34'11"	15°	15°25.7	.57811	.76105	.2833	.110	2120	3289	9.97 to 1
		CR. GEAR	70.010	70.381	69.393	4.1287	.1855	.3085	289	40°34'11"	15°	15°25.7	.57811	.76105	.2763	.123	2120	331	

FIGURE 18.

COMPARISON OF BATTLESHIP AND CRUISER GEARS.

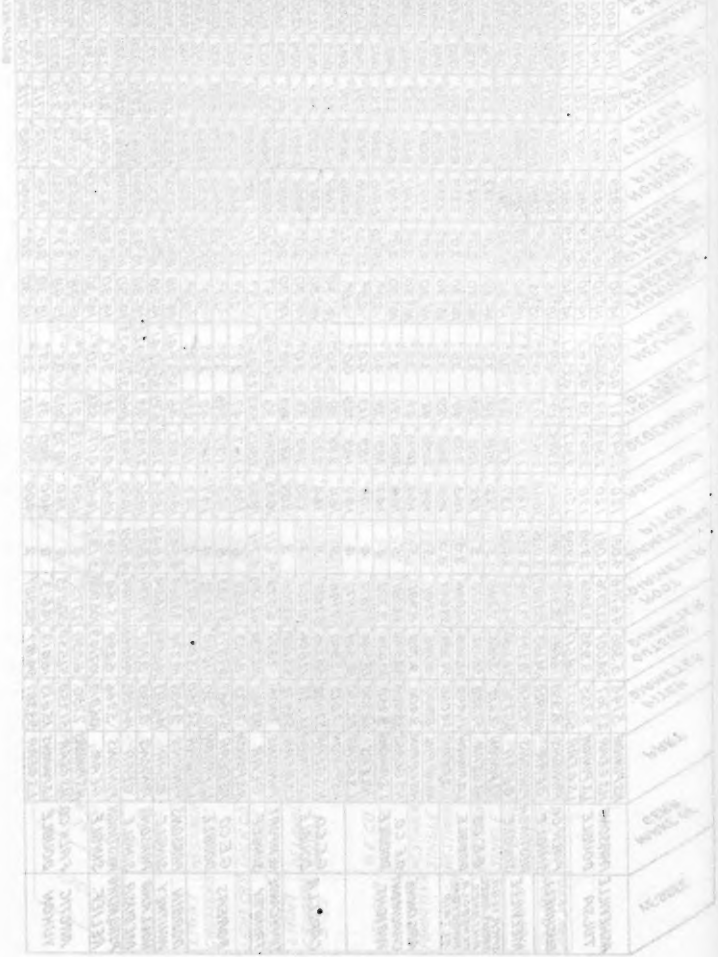
VESSEL	MAKE OF GEAR	PART	PITCH DIAMETER	OUTSIDE DIAMETER	ROOT DIAMETER	DIAMETRICAL PITCH	ADDENDUM	DEDENDUM	NUMBER OF TEETH	HELICAL ANGLE	NORMAL PRESSURE ANGLE	CIRCULAR PRESSURE ANGLE	NORMAL PITCH	CIRCULAR PITCH	THICKNESS OF TOOTH ON PITCH DIA.	ROOT CLEARANCE	S.H.P. TRANSMITTED	R.P.M.	REDUCTION RATIO
ASHEVILLE TULSA	PARSONS DOUBLE	H.S. PINION	5.146	5.386	4.6766	4.081	.120"	.2347	21	40°-45.9	14°30'	18°-50.5	.5829	.7697	.281"	.1147	400	3504	27.59 to 1
		H.S. GEAR	23.767	24.007	23.2976	4.081	.120"	.2347	97	40°-45.9	14°30'	18°-50.5	.5829	.7697	.281"	.1147	400	759	
		L.S. PINION	9.755	9.995	9.2856	3.998	.120"	.2347	39	42°-6.1'	14°30'	19°-12.9	.5829	.7857	.281"	.1147	400	759	
		L.S. GEAR	58.277	58.517	57.8076	3.998	.120"	.2347	233	42°-6.1'	14°30'	19°-12.9	.5829	.7857	.281"	.1147	800	127	
BUSHNELL	FALK CO. SINGLE	PINIONS	6.350	6.683	5.917	2.992	.1665	.2165	19	45°	18°-15'	25°	.7425	1.0501	.371"	.050"	1250	2053	16.42 to 1
		GEAR	103.820	104.153	103.387	3.005	.1665	.2165	312	45°	18°-15'	25°	.7391	1.0453	.369"	.050"	2500	125	
MELVILLE	WESTINGH. SINGLE	PINIONS	7.568	8.294	7.194	3.435	.363"	.177"	26	30°	14°30'	16°-38'	.7910	.9140	.393"	.040"	2000	1892	12.65 to 1
		GEAR	95.776	96.050	94.948	3.435	.137"	.474"	329	30°	14°30'	16°-38'	.7910	.9140	.383"	.051"	4000	110	
SPICA, VEGA, SIRIUS, RIGEL, ANTARES, ARCTURUS, PROCYON, ALTAIR	G.E. CO. DOUBLE	H.S. PINION	7.612	7.856	7.296	5	.122"	.158"	38	35°	16°-30'	20°	.5147	.6283	.253"	.036"	2500	3234	35.94 to 1
		H.S. GEAR	57.888	58.132	57.572	5	.122"	.158"	289	35°	16°-30'	20°	.5147	.6283	.251"	.036"	1250	425	
		L.S. PINION	11.442	11.768	11.026	3.5	.163"	.208"	40	35°	16°-30'	20°	.7353	.8976	.361"	.046"	1250	425	
		L.S. GEAR	54.058	54.384	53.642	3.5	.163"	.208"	189	35°	16°-30'	20°	.7353	.8976	.359"	.046"	2500	90	
ARGONNE CHAUMONT WRIGHT	G.E. CO. DOUBLE	H.S. UP PIN.	8.400	8.728	7.981	5	.164"	.2095	42	35°	16°-30'	20°	.5147	.6283	.252"	.0455	3000	3836	38.36 to 1
		H.S. L.P. PIN.	9.400	9.728	8.981	5	.164"	.2095	47	35°	16°-30'	20°	.5147	.6283	.252"	.0455	3000	3428	
		H.S. GEARS	38.200	38.528	37.781	5	.164"	.2095	191	35°	16°-30'	20°	.5147	.6283	.252"	.0455	3000	894	
		L.S. PINIONS	9.750	10.044	9.378	4	.147"	.189"	39	40°	20°	25°-25	.6016	.7854	.295"	.042"	3000	894	
		IDLERS	28.250	28.544	27.878	4	.147"	.189"	113	40°	20°	25°-25	.6016	.7854	.291"	.042"	1500	292	
		L.S. GEAR	82.250	82.544	81.878	4	.147"	.189"	329	40°	20°	25°-25	.6016	.7854	.291"	.042"	6000	100	
CAPELLA	G.E. CO. DOUBLE	H.S. PINION	7.612	7.940	7.146	5	.164"	.233"	38	35°	16°-30'	20°	.5147	.6283	.251"	.069"	2500	3380	32.56 to 1
		H.S. GEAR	57.888	58.216	57.422	5	.164"	.233"	289	35°	16°-30'	20°	.5147	.6283	.249"	.069"	1250	684	
		L.S. PINION	11.028	11.606	10.262	3	.289"	.383"	33	30°	17°-30	20°	.9092	1.0491	.447"	.094"	1250	684	
		L.S. GEAR	54.972	55.050	53.706	3	.289"	.383"	163	30°	17°-30	20°	.9092	1.0491	.445"	.094"	2500	90	
TIPPECANOE TRINITY	NEWPORT SINGLE	PINIONS	7.006	7.377	6.429	4.711	.1855	.2885	33	29°-35'-5	15°	17-10	.5799	.6669	.287"	.103"	1500	1454	16.16 to 1
		GEAR	113.161	113.532	112.584	4.711	.1855	.2885	533	29°-35'-5	15°	17-10	.5799	.6669	.287"	.103"	3000	90	
BOREAS	G.E. CO DOUBLE	H.S. PINION	7.833	8.043	7.559	6	.105"	.137"	47	27°	20°	22°-13'	.4665	.5236	.228"	.032"	3000	3375	37.5 to 1
		H.S. GEAR	57.667	57.877	57.393	6	.105"	.137"	346	27°	20°	22°-13'	.4665	.5236	.228"	.032"	1500	459	
		I.S. PINION	10.750	11.044	10.372	4	.147"	.189"	43	23°	20°	21°-45	.7230	.7854	.355"	.042"	1500	459	
		L.S. GEAR	54.750	55.044	54.372	4	.147"	.189"	219	23°	20°	21°-45	.7230	.7854	.355"	.042"	3000	90	
DOBBIN WHITNEY	PARSONS SINGLE	PINIONS	3.903	10.274	3.3264	4.645	.1855	.2883	46	32°-27-41"	14°-30'	16°-15	58302	67358	.2815"	.1028	3500	1550	14.76 to 1
		GEAR	146.183	146.552	145.604	4.645	.1855	.2888	679	32°-27-41"	14°-30'	16°-15	58302	67358	.2815"	.1033	7000	105	
HOLLAND MEDUSA	PARSONS SINGLE	PINIONS	9.330	9.701	8.754	4.609	.1855	.2883	43	31°-12-43	14°-30'	16°-39	58304	68677	.2815"	.1028	3500	1700	15.45 to 1
		GEAR	144.285	144.666	143.709	4.609	.1855	.2883	665	31°-12-43	14°-30'	16°-39	58304	68677	.2815"	.1028	7000	110	
PINGUINO RELIEF	WESTINGH.	PINIONS	5.966	6.290	5.164	5.197	.1665	.401"	31	30°	14°-30	16°-38	52358	6046	.261"	.2345	2650	1300	12.93 to 1
		GEAR	106.840	107.163	106.038	5.197	.1665	.401"	535	30°	14°-30	16°-38	52358	6046	.261"	.2345	5300	106	
ARCTIC YUKON	FALK CO. DOUBLE	H.S. PINIONS	7.750	8.050	7.375	4	.150"	.1875	31	45°	15°-48'	27°	.55535	.7854	.250"	.0375	1400	3580	40 to 1
		H.S. GEAR	51.250	51.550	50.875	4	.150"	.1875	205	45°	15°-48'	27°	.55535	.7854	.250"	.0375	1400	549	
		I.S. PINIONS	15.613	16.413	14.613	2	.400"	.500"	31	23°	18°-30'	20°	.1956	.1582	.712"	.100"	1400	549	
		L.S. GEAR	93.387	94.187	92.387	2	.400"	.500"	187	23°	18°-30'	20°	.1944	.1569	.712"	.100	2800	90	

*S.H.P. RELIEF 5000

FIGURE 19.

COMPARISON OF GEARS OF AUXILIARY VESSELS.

Figures 17, 18 and 19 show comparative data for all U. S. Naval vessels having reduction gears for main or cruising drive, with the exception of two or three of the older vessels which have been placed out of commission. These data are computed in accordance with Figure 16 and the accompanying definitions.



THE CARE AND PROTECTION AGAINST CORROSION OF SHIPS' HULLS AND FITTINGS.

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INTRODUCTION.

1. The deterioration of the hulls and fittings of steel ships is due primarily to corrosion. If corrosion could be completely eliminated practically all such parts would last indefinitely, as mechanical wear and tear, due to use and age, are negligible factors in determining the life of such parts. No means have, however, yet been found for permanently preventing the corrosion of the steel which enters into the construction of ships.

2. The expenditure of money in fighting rust and making good deterioration caused by rust is so great that the problem merits the most careful attention on the part of Naval personnel. At the outset it may be stated that there is no panacea for preventing corrosion. The only answer to the problem is hard work and intelligent effort in cleaning and painting surfaces, constant inspections to detect corrosion while it is still in the incipient stage, and the use of the most suitable protective coatings to meet the particular needs of each case. While the quality of the paint employed has much to do with the degree of protection afforded thereby, of even greater importance are the thoroughness with which metal surfaces are cleaned, and the freedom of the surfaces from rust, moisture, grease, etc., at the time of application of the paint. A poor paint applied to clean and bright metal surfaces will afford more protection than the best paint applied over rust, grease and moisture.

CHAPTER I—CAUSE OF CORROSION.

1. Steel corrodes only when it is exposed to both oxygen (air) and water. Steel exposed to an absolutely dry atmosphere does not corrode. Steel immersed in pure water from which all air

(oxygen) has been removed does not corrode. Under actual conditions, however, the atmosphere always contains moisture and water always contains air. Hence the problem of preventing corrosion, stated in its simplest terms, is to keep air and water from direct contact with the metal.

2. A number of theories have been advanced to account for the phenomenon of corrosion. The most generally accepted theory may be briefly summarized as follows: All metals have a tendency to dissolve in water, this tendency being known as the solution pressure of the metal. The solution pressure of iron is relatively high as compared to such metals as lead, silver, gold, platinum and some others. For this reason iron corrodes more rapidly than such metals. When an ion of iron dissolves in water an equivalent ion of hydrogen is set free. The rusting of iron (or of steel, which is the same thing so far as the chemistry of the reaction is concerned), may, therefore, be represented by the equation:

$$\text{Iron} + \text{water} = \text{hydroxide of iron} + \text{hydrogen}.$$

When this occurs the hydrogen ions spread out on the surface of the metal as an insulating film. If the iron and the water are both pure and the water is free of air the action ceases as soon as this occurs and no further reaction takes place. The metal is then said to be polarized. Quantitatively under such circumstances the amount of iron which goes into solution is so small that it is correct to say that iron (or steel) does not corrode in pure water free of air. These tendencies exist and these reactions take place whether the steel is actually immersed in water or whether it is merely exposed to the humidity of the atmosphere.

3. The chemical reaction under normal conditions does not, however, stop at this point. In the first place oxygen is always present. As soon as an ion of hydrogen is set free by the solution of an ion of iron it combines with the oxygen to form water, and the polarizing effect of the hydrogen ceases. More iron is then again free to dissolve, thus a continuous process is set up which is known as rusting. The first hydroxide of iron which forms is ferrous hydroxide. This combines further with oxygen to form ferric-hydroxide; this latter being what is ordinarily known as red

rust. Ferric-hydroxide may be still further oxidized to form the various oxides of iron familiarly recognized as scale.

4. In addition to the above there are many factors which either stimulate or tend to inhibit corrosion, the former being unfortunately the more common tendency. These factors will be briefly described:

(*a*) *Water and Dampness.*—In general it may be said that water and dampness stimulate corrosion. A practical illustration of this fact is that bilges, double bottoms, etc., which are kept dry suffer much less from corrosion than those which are usually wet or damp. Places that are alternately wet and dry suffer even more—for example, the waterline plating of ships.

(*b*) *Character of the Water or Atmosphere.*—Acid or acid gases, such as carbon dioxide and sulphur dioxide dissolved in water, increase corrosion. Harbor waters sometimes contain acids coming from the discharges from manufacturing plants, gas works, etc. Also in the vicinity of the outlets of sewers acidulated water will usually be encountered. Ships should, therefore, never be anchored or tied up near the discharge from sewers, gas works or other manufacturing plants employing acid processes. Water which is alkaline, that is, high in hydroxylions, inhibits corrosion; for this reason the water in boilers is maintained alkaline. The atmosphere of coal bunkers, especally at the bottom of the bunker, where dampness and water may also be present, stimulates corrosion because of the gases given off by the coal.

(*c*) *Mill Scale.*—The liberation of hydrogen and the formation of a hydrogen film on the steel when the metal dissolves has been mentioned above. Anything that prevents the formation of this film depolarizes the metal and, therefore, accelerates corrosion. This depolarizing action takes place with extreme ease on forge or mill scale—the magnetic oxide of iron which covers a piece of iron or steel after it has been highly heated. Thus, if this coat of scale is ruptured at any point corrosion at this point takes place with great rapidity and a pit is the result. For this reason mill scale should be removed with the greatest care, especially where the steel will be exposed to severe corrosive conditions, as in the bottom plating of ships. When a rivet is driven, sometimes, and

perhaps generally, hammer scale forms on the point of the rivet. Considerable pitting is often found on and around newly driven rivets on this account.

(*d*) *Non-Uniformity and Impurities in Steel.*—It is probable that steel which is uniform in quality and free of impurities corrodes to a lesser degree than non-uniform material. There is no very definite proof that impurities in steel stimulate corrosion, but it is likely that they have this effect because some of the impurities are in the nature of depolarizers and others have a difference of electric potential as compared to iron, thus setting up electrolytic action. This likelihood is one of the reasons for purchasing steel in accordance with specifications exacting a proper discard from the ingot from which the steel is rolled, as the upper part of the ingot contains most of the impurities.

(*e*) *Electrolysis.*—Stray electric currents, whether of internal or external origin, are frequently responsible for the corrosion of shell plating and portions of the internal structure of ships. Under the head of electrolysis may be placed the violent corrosion resulting from lying alongside of copper-sheathed craft or piles, and the serious local corrosion of inner bottom plating and framing sometimes encountered due to permitting brass filings and pieces of brass or copper pipe to lie on inner bottom plating and in bilges. As the bilges and inner bottom plating of machinery spaces are frequently covered with water, it is particularly important that scrap copper and brass should not be allowed to lie in such locations due to the electrolytic action which takes place between the copper and the steel. Ships have been known at times to make overboard connections from switchboards with the sea in order to improve the lighting. This practice always results in serious pitting of the shell plating. Frequently, also, ships lying alongside of docks experience violent pitting due to stray electric currents from submarine transmission lines.

(*f*) *Zinc Protectors.*—Zinc is electropositive to iron. Therefore, when zinc and iron are placed in contact with each other and are immersed in water the zinc corrodes rapidly and the iron does not, the theory to account for this action being that the hydrogen equivalent of the zinc which dissolves, plates out on the iron, thus protecting it against corrosion. For this reason readily renewable

zinc plates and rings are fitted to the underwater body of ships in the locations where corrosion is particularly violent—around certain valve openings, on rudders, propeller struts, etc. In recent years experiments have been made which indicate that soft steel protectors are equally efficacious in preventing corrosion in many such locations, and are cheaper to maintain.

CHAPTER II—PROTECTIVE COATINGS.

1. As mentioned above, corrosion of steel can be prevented by keeping the metal from contact with air and moisture. In practice this is difficult of accomplishment even under the most favorable circumstances and quite impossible under many circumstances. However, the manner in which all protective measures function is to prevent this contact, either permanently, or for shorter periods of time, by renewing the protective coating when it begins to break down. Protective coatings may be broadly divided into two classes, as follows:—

(*a*) Coatings consisting of some other metal more highly resistant to corrosion than steel, such as zinc, tin, lead, nickel, silver, copper, porcelain enamel, etc. This method is used to a considerable extent for protecting fittings and equipment used on board ship. For example, galvanized (zinc coated) stanchions, dogs, cleats, light plating, etc.; lead-lined salt water piping; tinned condenser tubes, sheet metal, etc.; nickel or silver plated table ware; enameled bath tubs, etc.

(*b*) Coatings of paint, varnish, bituminous compounds and other liquid compositions which dry, after application, to a tough elastic film. The preservation of the ship structure by applying protective coatings of this kind is the only method which can be practiced by the personnel afloat.

2. *Liquid Protective Coatings.*—There is no unanimity of opinion on all the details connected with the subject of protecting steel by liquid coatings, the best ingredients to use in their manufacture, their suitability for various purposes, etc. The financial interests of manufacturers of paints and varnishes and their ingredients have much to do with the claims made for the various products. There are, however, a few fundamental principles about which there are no uncertainties, as follows:—

(*a*) No one kind of paint or varnish is suitable for all locations. For example, for the exposed upper works of ships oil paints are the most suitable from the point of view of durability, appearance and ease of repainting. On the other hand, for the bilges of engine and boiler rooms, which are more or less continually wet, bituminous compositions are more suitable than oil paints. The question of cost also enters in determining the suitability of a paint for a particular purpose. A cheap paint may be good enough for use in certain places but very uneconomical in other places. For instance, a cheap asphaltum is good enough for use on anchors, anchor chains, chafing plates under chain cables, etc., because in any case the paint in such places wears off mechanically before it deteriorates from age. However, for the bilges of a ship, where mechanical wear is not a factor, asphaltum paint would be very uneconomical. A coating is needed in such places which will resist self-deterioration as long as possible.

(*b*) The protective film must be sufficiently elastic to expand and contract with changes in temperature, but must still be hard enough to resist, as far as possible, mechanical abrasion and softening from water or moisture.

(*c*) The protective coating must not contain an ingredient which will attack the steel. If such ingredients must be added to serve the purpose for which the paint is to be used, as for example, to prevent the fouling of a ship's bottom, an insulating coat of neutral paint must first be applied.

3. *Paint.*—Broadly speaking, it may be said that paint consists of two parts, namely, the pigment and the vehicle which carries the pigment. The latter undergoes a change to the solid state when the paint dries, or some of the ingredients of the vehicle, such as turpentine, naphtha, etc., evaporate. These two must form, in drying, a firm, impervious coating on the surface of the metal, but must not become so hard as to be inelastic and brittle.

4. *Linseed Oil.*—Linseed oil is the most commonly used ingredient of paint vehicles. This oil is what is known as a "drying" oil, that is to say, an oil which, when exposed to the atmosphere, changes from a liquid to a solid, elastic and leathery substance. This action takes place not by evaporation but by oxidation whereby the oil absorbs oxygen to the extent of from 10 to 18 percent

of its weight. The solid which is formed is called linoxyn. It is really in the linoxyn that most of the merit of the paint film lies. Anything which prevents the formation of this tough, elastic film of linoxyn is detrimental to the paint as a protective coating. It is for this reason that one coat of paint should always be allowed to dry thoroughly before the next is applied, otherwise the oxygen does not have access to the underlying film, which, therefore, does not become as hard as it should be. Also for this reason adulterants in the paint, such as non-drying oils, are objectionable. Linseed oil in the raw state requires a considerable length of time to dry. To hasten the drying certain metallic oxides or salts (driers) are incorporated in the oil by heat or otherwise. The subject of driers will be discussed further on, but it may be mentioned at this point that the indiscriminate and excessive use of driers in paints, for application to steel, is objectionable because by their very nature as oxidizing agents they may attack the steel and cause the very corrosion which it is the object of the paint to prevent. Also, the excessive use of driers hastens by oxidation the destruction of the paint film.

5. The quality of the linseed oil used in making a paint is, in a large measure, the criterion of the quality of the paint itself. Linseed oil is extracted from the seed of the flax plant. Formerly the seed was pressed cold in order to prevent, as far as possible, the inclusion of solid fats, fatty acids, and the juices from other vegetable matter mixed with the flax seed, as these ingredients are detrimental to the drying of the oil. In order to obtain the maximum yield of oil from the seed, manufacturers began the practice of pressing the seed while hot. Commercially a distinction was formerly made, and is still made in some regions, between cold-pressed and hot-pressed linseed oil, the former bringing a higher price because of its superior quality. In recent years filtering processes have been developed for removing, to a great extent, the detrimental solid fats, fatty acids, etc., from hot-pressed oil. Furthermore, by keeping the oil in storage in large vats for a considerable time these detrimental substances settle out in a large measure. Specifications for linseed oil should be drawn so as to detect and reject excessive percentages of these substances. Nevertheless these fatty non-drying substances are present to some extent even

in linseed oil of good commercial grade. As the oil ages and stands they continue to settle to the bottom. For this reason linseed oil should always be drawn from the top of the barrel or container without shaking or stirring. The sediment at the bottom of the container should never be used in making paint.

6. There are a few other oils which are suitable, in varying degrees, for mixing paints and varnishes, but in general, with the exception of china-wood oil, which has particular merit for making waterproof varnishes, there is no other oil as suitable for making paints for marine work as linseed oil. Cheaper oils of semi-drying character, such as cotton-seed oil, are frequently used as adulterants for linseed oil. Their use is objectionable, except in special cases, as they do not harden completely.

7. *Thinners.* (*a*) A paint made by mixing only linseed oil and pigments is too thick to work out smoothly under the brush. The resulting paint film is likely to be streaky, uneven, and frequently leaves spots imperfectly covered. To facilitate the mechanical process of spreading the paint evenly, thinners are added so as to make the paint more fluid. Thinners are volatile liquids which form homogeneous solutions with oils or resins and on exposure to air evaporate more or less completely. The principal thinners used in the manufacture of paint and varnish are the following:—

Turpentine (pure gum spirits).
Wood Turpentine.
Mineral Spirits (turpentine substitutes).
Benzol.
Toluol.
Coal-tar Naptha.
Alcohol.

(*b*) Turpentine (gum spirits) is derived by distillation from the sap of certain species of pine trees. Wood turpentine is made by distilling pine wood, stumps and roots. Mineral spirits is one of the light volatile products obtained from the distillation of crude oil. Mineral spirits is much cheaper than turpentine, and, provided it has the proper physical and chemical properties, is equally satisfactory as a thinner. It has practically replaced turpentine as a paint thinner.

8. Thinners are added to paint for their mechanical effect alone; to facilitate spreading, to reduce excessive proportions of oil, to hasten the setting of the paint, to assist penetration of the first coat on wood or other porous surfaces, to reduce the gloss of undercoats so as to improve the adhesion of subsequent coats, and to give a flat (dull) finish where desired.

9. Thinners should not be used in excess, as they impair the durability of the paint by reducing the proportion of oil. It should always be borne in mind that the oil is essential to the life of the paint film. If the oil is thinned excessively the paint will have less binding power, the film will be thinner, and consequently will be less durable.

10. Benzol, toluol and coal-tar naphtha are products of the distillation of coal-tar. They are good solvents and are used extensively in paint and varnish removers. As they penetrate wood readily, they are used in oil stains and sometimes as additional constituents of the vehicle for the first coat of paints on resinous wood. They are also largely used to thin various bituminous compositions. As they are good solvents and evaporate rapidly, thereby accelerating drying, they are used in some types of ships' bottom paints. As a general rule these liquids should not be used as thinners for linseed oil paints, as they evaporate too rapidly.

11. In making protective coatings, alcohol is used principally as a solvent for shellac. In ship bottom paints of the shellac type alcohol is the chief ingredient of the vehicle. Alcohol is not a suitable thinner for linseed oil paint.

12. *Driers.*—As previously mentioned the chemical reaction which takes place when linseed oil dries consists of a combination of oxygen with the oil to form a hard elastic substance called linoxyn. There are other oils which have the property of absorbing oxygen, but to a much smaller degree. If an oil has this property, but only to a slight degree, it is called a semi-drying oil. Oils which do not have this property are called non-drying oils. Even in the case of linseed oil the oxygen absorption process is slow. To accelerate the process compounds are added, which, in common parlance, are called driers and which technically are known as catalysers. A dryer or catalyser accelerates the drying process by conveying oxygen from the air to the oil. The dryer during this

process temporarily changes its composition but as soon as it has given up oxygen it resumes its original condition and is then in a position to repeat the process. Driers are intended to act only on the oil in a paint or varnish and not on the pigment.

13. Various kinds of driers are employed, depending on the nature of the paint or varnish with which they are to be used. The most active driers are certain oxides of lead, manganese, and cobalt. Commercially, driers are usually furnished dissolved in a mixture of linseed oil and volatile oils such as turpentine and mineral spirits. A liquid dryer of this character, which, in addition, contains gums, is called a japan dryer. The final destruction of a paint film is due to oxidation, in other words, burning, unless the film is sooner destroyed by mechanical wear, corrosion underneath, moisture or extraneous causes. The addition of an excessive amount of dryer to the paint will therefore hasten the complete oxidation of the film. Too much dryer may also cause darkening of the paint, in much the same manner as a piece of white cloth is darkened or singed when it is pressed with an iron that is very hot.

14. *Pigments.*—A great variety of pigments is used for making paints. Only those which are most commonly employed for marine work will be described.

15. *Red-Lead.*—It is generally conceded that there is no better priming coat for steel than red-lead paint. The superiority of red-lead is due to the fact that this compound has a very high attraction for linseed oil as compared to other pigments. The incorporation of the particles of this pigment with the oil is therefore more intimate than in the case of other pigments, which results in the formation of a paint film of great durability. Red-lead is made by roasting litharge in contact with air which, thus converting the latter (PbO) into the former (Pb_3O_4). That portion of the litharge which is in the form of a fine dust is easily oxidized but the coarse particles, while they become red on the outside, still consist of unchanged litharge on the inside. Even prolonged roasting fails to convert the coarser particles into Pb_3O_4. The essential preliminary step to converting all of the litharge into red-lead is extremely fine grinding. This is expensive and for that reason red-lead which contains only a small percentage of litharge costs

more than red-lead containing 15 per cent of litharge,—the latter being approximately the percentage in the more common commercial grades of red-lead.

16. Red-lead containing only a very small percentage of litharge,—less than 2 per cent, has several advantages over the grades containing considerable litharge. In the first place a very finely ground pigment makes a more durable paint than a coarser pigment. Red-lead which is practically free from litharge must necessarily be finely ground. In the second place litharge acts on linseed oil at ordinary temperatures, the paint becoming viscid and ropy if not used promptly, and finally, if allowed to stand, becomes a hard cement like solid. Red-lead, however, is such an excellent paint for the priming coat on steel that even though the grades low in litharge cannot be obtained, it is better than other pigments for most purposes. Hardening due to the presence of a considerable percentage of litharge can be largely circumvented by mixing only enough paint at a time to cover the needs for a few days in advance. This is, however, not always convenient. For that reason an inert compound, such as silica, is often mixed with the red-lead. This acts mechanically to prevent the agglomeration of the particles of red-lead. The durability of the paint is somewhat impaired however by the addition of inert materials of this kind.

17. Red-lead can be most conveniently handled in paste form, that is, ground in linseed oil, but only the grades containing less than 2 per cent of litharge will, even when sealed up in cans, remain soft in this form for any length of time. The advantage of using red-lead in paste form are several,—the paste can be more readily incorporated with the oil in making paint and the danger from lead poisoning, to those handling the material, is less than in the case of dry red-lead.

18. Red-lead is not generally used as the final coat of paint because of its color, (although by the addition of a small amount of lamp-black it is often used as a finishing coat where a black color is satisfactory), because of its cost, and, because it discolors with age.

19. *White-Lead.*—White-lead is a basic carbonate of lead. It is a valuable and much used pigment, both for wood work and for finishing coats on metal. It has great opacity and good working

qualities, but it is subject to rather rapid deterioration when used alone, due to chalking. White-lead, by itself, also darkens with age, especially where exposed to a sulphurous atmosphere. Basic sulphurate of lead also has considerable application as a white pigment. White-lead is seldom used by itself as a pigment but is much used in connection with zinc oxide.

20. *Zinc Oxide.*—This pigment is the whitest of the various pigments used for making white paint. It has, however, a tendency, when used alone in making an oil paint, to check and to scale because of its extreme hardness. In combination with white-lead it makes the most durable of the white paints as the disabilities of the two pigments offset each other. It is unaffected in color by any gases present in the atmosphere, has no effect on any pigment with which it may be mixed, and is non-poisonous. Zinc oxide is used in enamel paints by itself without the addition of white-lead, as the varnish in such paints adds an element of toughness which to a considerable extent prevents a pigment even as hard as zinc oxide, from checking and scaling. As pure zinc oxide is an expensive pigment, there are numerous brands on the market which contain a certain percentage of lead sulphate and zinc sulphate. Zinc oxide containing these compounds can be made by using cheaper grade ores than the material used in making pure zinc oxide. For many purposes leaded zinc oxides of this kind are perfectly suitable and save money.

21. *Iron Oxides.*—Iron oxides are used to a considerable extent in making paint. These oxides are cheaper than red-lead. Paints made therefrom do not have the durability and toughness of red-lead and cannot, from the point of view of protecting steel against corrosion, be considered an equally satisfactory substitute for red-lead. For naval work they are used principally to give the desired color to a finishing coat of brown or the various shades of red.

22. *Extenders.*—An extender is a white or colorless substance added to white or colored paints. Extenders are sometimes used to form the solid base in which staining colors or dyes are precipitated and to decrease the preponderance of chemically active pigments in the paint film. Some of the extenders in common use are:

Barium sulphate (Barytes, blanc fixe, permanent white).
Silica (silex, silicious earth).
Magnesium silicate (asbestos, asbestine, pulp talc).
Aluminum silicate (china clay, kaolin).
Calcium sulphate (gypsum, terra alba).
Calcium carbonate (white mineral primer, Paris white, whiting, etc.).

23. Extenders are inert in the sense that they are chemically stable; that is, they do not change their composition when exposed to the atmosphere, as for example, white-lead does; neither do they act upon or are they acted upon by any chemical constituents in the paint. By themselves these compounds are not suitable for making paint. As compared to white-lead, red-lead and zinc oxide, they have a low attraction for linseed oil which is probably the principal reason why they do not form a tough and durable paint film. If used at all in making paint these compounds should be used sparingly and with discretion.

24. *Color Pigments.*—Color pigments are used in conjunction with white base pigments, to produce any desired shade of color. They are also used alone with the necessary vehicles. They may be divided into the following classes, based on the methods of manufacture, viz.:

Natural Earth Colors.—These are found as deposits in the earth and utilized as pigment either in their natural state, after grinding and purification, or after further treatment, such as oxidation by burning, calcination, etc. Some of the colors are: Indian red, ochre, metallic brown, siennas, umbers, mineral blacks.

Chemical Colors.—Chemical colors are pigments produced by chemical action of one substance, usually in solution, upon another substance, resulting in the formation of a colored compound. Some of the chemical colors are: Prussian and Chinese blue, lead chromate, chrome green, ultramarine blue, cobalt blue, vermillion, etc.

Carbon Blacks.—The carbon blacks are practically pure carbon. They comprise (*a*) lampblack, which is a specially prepared soot from oil lamps; (*b*) gas blacks, from natural gas; (*c*) graphite, which was originally a natural product, but which now may also be produced by means of the electric furnace; (*d*) bone black, ivory black, drip black, vine black, etc., made by carbonizing animal

and vegetable substances. The carbon blacks have great covering capacity in proportion to their weight. Their durability and tinting power is good, but on account of their electrical conductivity, tending to stimulate corrosion, they should never be used in priming coats on steel. In any case, they are seldom used alone as pure colors. It is better to grind with them such a proportion of colorless inert pigments as will measurably increase the thickness of the paint film without impairing its quality.

25. *Varnish.*—A varnish is a fluid, usually transparent or translucent, though sometimes opaque, which when spread upon a surface in a thin film dries by the evaporation of its volatile constituents and the oxidization of the other constituents, or by evaporation alone. Oil varnishes are solutions effected by the heating of gum resins and drying oils with the addition of a small amount of metallic salt to facilitate drying, and with sufficient thinners to obtain the desired fluidity. Spirit varnishes are solutions of gum resins and volatile liquids.

26. Varnish is used principally as a protective and decorative coating for wood. It also forms part of the vehicle in enamel and gloss paints, such as are used for the final coat of white paint on interior work. Enamel painted surfaces can be kept clean more readily than oil painted surfaces because dirt and grease do not adhere so firmly to the smooth and glossy surface characteristic of enamel paints. Such surfaces can be washed without injuring the coating to the same extent as would be the case with an all oil paint. Where appearance and durability are of importance a varnish made of fossil gums and china wood oil should be used. Such varnishes are more resistant to the softening influence of water than oil paints or the cheaper grades of varnish. For this reason such varnishes are often used for exterior work where a bright finish is desired for the sake of appearance, as for example, on the upper works of high grade launches, gangway ladders, etc. Varnishes for exterior work are called spar varnishes. Varnish is also employed as the vehicle for boottopping for use at the water line of ships, as it dries more quickly than oil paint and resists the conditions of alternate dryness and wetness more effectively than oil paint. Good varnish is, however, more expensive than paint, therefore this fact must be taken into consideration in deciding on its use as a substitute for paint.

27. *Bituminous Compositions.*—These compositions are encountered under many different trade names, but are all fundamentally alike. Bituminous composition may be defined as a bituminous solution which is applied cold as a priming coat and a bituminous enamel which is applied hot over the solution. The latter, on cooling, hardens and forms an enamel-like surface. Bituminous enamel differs essentially from such protective coatings as paint and varnish in that the former is not dependent on the oxidation of oil for the formation of the protective film.

28. Bituminous enamel does not soften under the influence of water to the same extent as an oil paint. If properly applied over a clean surface, and if the material is of good quality, there is little doubt that bituminous compositions provide better protection than oil paints in such places as machinery space bilges, the top of inner bottom plating, the inside of tanks used for cold water, etc. Such composition should not however be used in feed water tanks because hot water eventually softens the enamel which then sooner or later finds its way into the suction piping. There are several objectionable features connected with its general use. In the first place, unless the enamel is heated to just the right temperature and applied at the right temperature, it is brittle after cooling. With temperature changes of the metal underneath, this lack of elasticity may cause the enamel to crack. If too hard the enamel also chips off from mechanical injury. A further objection is its black color, which makes detection of corrosion and dirt difficult. Corroded spots can also not be touched up locally as readily as in the case of a surface which is painted. A further objection to bituminous composition is the difficulty of applying the enamel in confined spaces, such as double bottoms and small tanks, because of the fumes which the material gives off when it is hot.

29. *Portland Cement and Tiling.*—A wash made of Portland cement and fire clay is fairly satisfactory for fresh water tanks, as such a coating does not impart an unpleasant taste to the water. Before the development of bituminous compositions all compartments and tanks, occasionally or frequently used for water, were protected by cement wash. A coating of this kind has however, no elasticity and loosens from the metal very easily when dry. For

this reason bituminous compositions are now used largely for such spaces, except for feed water tanks, where there is danger that hot water will soften the bituminous composition. Cement wash is the cheapest of all protective coatings, but as already mentioned it has no merit except in protecting the interior of tanks which are constantly wet. Even then it must be carefully watched and frequently renewed to prevent corrosion from starting under the cement.

30. The most suitable protective coating for decks of water-closets, bath rooms, galleys, etc., is ceramic tiling laid in cement. Tiling is also frequently used for the deck covering of dynamo rooms, condenser rooms, etc. One of the principal objections to tiling is its weight. Appearance alone is not a sufficient reason for using tiling as a deck covering because of the objectionable feature of excessive weight. Its use is justified only in spaces where the decks are more or less constantly wet, where there is difficulty in keeping the deck clean or where appearance is of major importance.

31. *Ship Bottom Paints.*—The object in applying paint to the bottom of ships is twofold. First, to prevent fouling of the bottom by marine growths, and second, to prevent corrosion of the steel plating. A foul bottom increases the resistance which the ship encounters in passing through the water; hence it reduces the speed and increases the consumption of fuel. Many brands of paint are on the market for preventing the fouling of bottoms. All brands contain poisons of one kind or another which are intended to kill marine growths or to prevent their attachment and hence to prevent fouling. No anti-fouling paint will indefinitely prevent fouling because the poisons sooner or later dissolve out of the paint film, or the film disintegrates.

32. One of the requisites of ship bottom paints is quick drying, because ships cannot usually be kept in dry dock for any length of time. For this reason oil paints are not suitable for this purpose as all oil paints require considerable time if they are to dry thoroughly. Sometimes when very serious corrosion and pitting has started in the shell plating, the application of a coat of red-lead, after thorough and complete scaling, will check the corrosion provided that the red-lead is given at least four weeks to dry be-

fore applying the anti-fouling paint and undocking. If as much as four weeks are not available for drying it is better to apply a coat of anti-corrosive paint.

33. Anti-fouling paint must never be applied directly to the steel hull because the paint has insufficient anti-corrosive properties, permitting rust to form which destroys the paint film. It is therefore necessary first, to apply a coat of other paint, called an anti-corrosive paint, to act as an insulator between the anti-fouling paint and the steel. The vehicle of all anti-corrosive ship bottom paints consists of a liquid which evaporates quickly and completely, such as alcohol or one of the derivatives of coal-tar, such as naphtha. Anti-corrosive paint which is suitable for use with one brand of anti-fouling paint may not be suitable for another because of differences in the vehicles. For example, an anti-fouling paint made up with an alcohol shellac vehicle will adhere firmly only over an anti-corrosive paint which is also made with an alcohol shellac vehicle. In the same way a naphtha vehicle anti-fouling paint will give the best results only over a naphtha vehicle anti-corrosive paint. While the first coat of ship bottom paint is called anti-corrosive paint, for purposes of identification, such paint has no merit for preventing corrosion as compared to oil paints in any other location than on the ship's bottom. Due to the fact that the vehicle must necessarily be one which evaporates completely, permitting the paint to dry quickly, the film of anti-corrosive paint left on the metal can have little toughness and durability as compared to the film resulting from the use of a linseed oil paint.

34. There are several types of anti-fouling paint and many brands under each type. One type consists of the liquid vehicle kind which is applied cold and which dries due to the evaporation of the vehicle. To this type belong the most commonly used brands. The vehicle is similar to that used in all anti-corrosive paints, namely, either alcohol or one of the coal-tar derivatives like naphtha. The other type may be broadly classified as the soap type. This kind of paint must be heated to reduce it to the liquid form. It solidifies on cooling, but never becomes very hard. The latter type is considerably more expensive both as to first cost and as to cost of application, but has considerable merit in preventing fouling.

CHAPTER III—PREPARATION OF SURFACES.

1. No matter how excellent the quality of the protective coating may be, it will not afford protection against corrosion unless the surface to which it is applied has been properly prepared. A clean surface, free from rust, grease and moisture, is absolutely essential to the protection of steel by paint or other protective coatings. It is better not to paint at all than to paint over rust and scale, because under the latter circumstances the actual condition of the metal is temporarily hidden by the paint, thus contributing to further neglect.

2. *Removal of Mill Scale.*—Mention has been made of the fact that mill scale stimulates corrosion. For this reason mill scale should be removed before steel is worked into the structure of a ship, especially when the location is such that the steel will be exposed to water or considerable dampness. In general, mill scale should be removed from all outside plating below the line of the deck next above the water line; the inner bottom plating throughout; all plates, brackets and shapes in the whole of the interior of such double bottoms or other compartments as are intended for oil or water tanks; the last strake of all bulkhead plating in or bounding the machinery spaces; and all structural plates and shapes rendered inaccessible for inspection and painting by casings or ceilings which, because of wiring leads or other construction, cannot be provided with practicable portable portions.

3. There are several methods of removing mill scale. The most satisfactory method is pickling. In this process the steel is immersed in a bath of either dilute hydrochloric or dilute sulphuric acid a sufficient length of time to completely remove or soften the mill scale. If hydrochloric acid is used the proportion should be one volume of acid to nineteen volumes of water. In the case of sulphuric acid, one volume of acid to forty volumes of water is the proper strength. The decision as to which acid to use depends more on the relative cost of the two acids than on any other consideration. A sulphuric acid bath of the concentration stated gives somewhat quicker results than hydrochloric acid. The length of time required for removing the scale depends upon the thickness and newness of the scale. The scale will usually come off in from

eight to ten hours in solutions of the strengths given. If the steel has just come from the mill more time may be necessary. If the material has been exposed to the weather several months not so much time may be needed. The action of the acid can be accelerated by heating the pickling solution. In order to prevent pitting care must always be exercised in pickling not to leave the steel immersed too long. When the steel has been in the pickling bath a sufficient length of time to remove or loosen all of the scale it should be taken out and washed with fresh water by means of a hose, or it should be dipped in a fresh water tank. Any remaining scale should then be removed by wire brushing. The steel should next be dipped in a bath of lime water to neutralize the adhering acid. After removal from the lime water the steel should again be thoroughly washed with fresh water and allowed to dry before being put to use.

4. Mill scale can also be removed by sand-blasting, but this is a more expensive process than pickling once the pickling tank equipment is available. Mill scale gradually loosens when exposed to the weather. The scale on plates and shapes that have been exposed six months or more is often loose enough to permit removal by vigorous wire brushing and hammering. Generally, however, spots remain with very tightly adhering scale. Such spots can be cleaned effectively by local heating with a torch, but this practice is not advisable in the case of thin plates because of the risk of buckling or setting up strains in the metal.

5. *Ships' Bottoms.*—Marine growths can best be removed from the bottoms of ships by means of scrapers. This can be done most cheaply and quickly by working with long-handled scrapers from boats or floats as the water recedes in docking. Such growths are easy to remove when wet, but difficult to remove if allowed to become dry. At best, however, the cleaning which can be accomplished while the water is receding is not sufficient unless the dock is pumped down very slowly, hence additional cleaning with scrapers is usually necessary afterward.

6. As soon as the dock is dry the bottom of the ship should be carefully inspected for scale, pitting and any other extensive corrosion. Scale can be removed only by means of chipping hammers. Hand hammers are better than pneumatic hammers for

this purpose unless the men have been properly trained in the use of the latter type of hammers. An inexperienced man is likely to bear down too hard on a pneumatic hammer, which results in cutting the metal. This is not so likely to happen with hand hammers. A steel surface covered with fine cuts such as result from the improper use of pneumatic hammers will corrode more rapidly than a smooth surface. If a large area must be scaled, and if pneumatic hammers are available, men should be trained in their use, as the work can be done more rapidly with such equipment. A pneumatic hammer should, however, never be turned over to a man without giving him instructions in its use.

7. The tendency in chipping a ship's bottom or other extensive surface is to scatter effort over a considerable area, leaving islands of untouched scale and rust. The procedure which gives the best results is to assign a specific area of the bottom, for cleaning and scaling, to each man. The assignments may be made each day or for the entire job. The assigned tasks should be inspected to see that the work has been thoroughly done before men are moved to other areas. The men should be taught to scale inch by inch and not by hammering in one place a few moments and then beginning to hammer some other place perhaps six inches or a foot away. In laying out the job the men should not work too close to each other, as this results in interference, one with the other. Also, unless the men are properly spaced it is difficult to assign a definite area to each man and to inspect his work. Just before painting the surface should be wire-brushed to remove any loose scale or rust. The wire-brushing should be done just ahead of the painters so that no area will stand more than a few hours after being wire-brushed before it is covered with a coat of paint.

8. Old paint which adheres firmly with no evidence of scale underneath should never be removed. This is especially true in the case of ships' bottoms as their painting for appearance is never a consideration. If the mill scale has been properly removed from bottom plating, with proper protection of the plating during construction, and if thereafter the ship is docked frequently during the first two or three years of its life, a layer of paint is often built up which becomes as firm as enamel. Such a layer of paint is extremely valuable and should never be disturbed.

9. *Removal of Oil and Grease.*—No kind of paint will adhere to a greasy or oily surface. Before painting, oil or grease should, therefore, always be wiped off. At times it may be found necessary to resort to the use of gasoline to accomplish this effectively. Grease and oil are especially likely to be encountered on the hull plating of ships along the waterline and in the wake of scuppers.

10. *Preparation of Galvanized Surfaces for Painting.*—In the process of galvanizing a film remains on the surface to which paint does not adhere firmly. Before painting this film should be removed by cleaning with ammonia or vinegar. A still better priming liquid consists of two ounces of nitrate of copper, two ounces of copper chloride and two ounces of ammonium chloride, dissolved in one gallon of water to which two ounces of hydrochloric acid should be added after the chemicals have dissolved. This solution should be painted on the galvanized surface with a brush, allowed to dry, and then dusted off just before applying the first coat of paint.

11. *Preparation of Wood Surfaces.*—A satisfactory job of painting cannot be done on wood unless the surface is smooth. Planing is generally sufficient, but at times rough spots remain which must be sandpapered. All knots should be brushed with a thin coat of shellac before applying the first coat of paint. This treatment binds in the excess resin in the knot, preventing its striking through subsequent coats of paint. At least 15 per cent of turpentine or mineral spirits should be used in the priming coat to secure penetration. Before applying a second coat, all nail holes and crevices should be filled with putty.

12. *Preparation of Surfaces in General.*—Scaling and cleaning, preparatory to repainting, is at best uninteresting work. If left to enlisted men or yard workmen without constant and intelligent supervision, it is certain to be slighted and to be done improperly. The personnel engaged on such work must, first of all, be instructed. While no difficult technique is involved in acquiring proficiency in such work, there are a sufficient number of fundamental requirements to warrant the time necessary to instruct all men and gang leaders in their work. The few fundamentals as to preparing surfaces have already been mentioned under previous headings. These instructions should be given by officers to petty

officers and gang leaders as well as to enlisted men and workmen from time to time. Gang leaders and petty officers should be held responsible for the character of the work done by the men, but the daily inspection of the work accomplished must be done by officers, otherwise it is certain to be slighted. It is not enough for an officer to inspect the work occasionally or when the whole job is completed, as, for example, the scaling, cleaning and painting of trimming tanks or double bottoms. The more difficult the access to the space on which work is being done the greater is the need for supervision and inspection during the progress of the work. Constant inspection and instruction of the personnel engaged on the work is necessary. As ships' crews and yard workmen are changing continually, the instruction of personnel engaged on such work can never be considered as completed.

13. A large proportion of the painting done on board ship does not require scaling or removal of the old paint. Old paint which adheres firmly and which shows no signs of rust underneath should never be removed. Before repainting a surface of this kind it should be scrubbed with soap and water. The surface should then be rinsed off with clear water. Care should be taken not to permit the soapy water to run down on other painted surfaces without immediately rinsing off. Where the old paint is lumpy or thick, fine sand may be used with the soap, but under no circumstances should steel brushes or scrapers be used unless necessitated by the presence of rust, or unless the whole surface is to be scraped down to the metal. Care should be taken to insure that the surfaces are thoroughly dry before paint is applied. Magazines, storerooms and spaces little frequented should be repainted for protection only and not for appearance. In such cases, if there is no evidence of grease or dirt on the old paint, scrubbing with soap and water may be dispensed with, but even in such cases the surface should be brushed off and all dust and dirt carefully swept up at the bounding angles along the deck, so as to insure that the new paint will not be applied over dust and dirt.

14. In view of the extreme thinness of bulkheads, shell and other plating on destroyers, and as such vessels are built partly of galvanized material, the galvanizing of which may be damaged by the use of scaling hammers, old paint and rust on structural work

or fittings of such vessels shall be removed by means of scrapers and wire brushes. Scaling hammers shall be used only where actual scale exists which cannot be removed otherwise.

CHAPTER IV.—THE PRACTICAL APPLICATION OF PROTECTIVE COATINGS.

1. *Mixing Paints.*—The paints which are used in the largest quantity on board ship are issued ready mixed. As paint is a mixture of solid pigments with a liquid vehicle, the former will in time settle to the bottom of the container. Separation of this kind takes place sooner in some paints than in others, depending on the pigment, but there is always some separation of the pigment within a short period of time after manufacture. In the case of heavy pigments, such as red-lead and white-lead, a solid mass may even form in the bottom of the container if the paint is at all old. Paint should, therefore, never be used until all the sediment in the bottom of the container has been stirred up and mixed with the vehicle. If the paint is used without mixing, it is obvious that the upper portion will contain too much oil and too little pigment. It should always be remembered that both the pigment and the oil in the paint are necessary to form a durable protective film.

2. An oar-shaped paddle is the best implement for mixing paints by hand. A jerky motion of the paddle will give better results than a slow motion. Paints issued in ten-gallon containers should be mixed by inserting the paddle through the filling hole, as this provides better leverage for manipulating the paddle. Paint delivered in small containers should be mixed with a stick. A paint brush should never be relied on for mixing the paint, as it is bad for the brush and the mixing cannot be thoroughly accomplished.

3. At least fifteen minutes should be devoted to energetic stirring before a ready-mixed paint is applied. It is particularly important to thoroughly mix anti-fouling paint, as the poisons are heavy and settle to the bottom of the container. If not thoroughly incorporated and uniformly distributed throughout the vehicle the poisons will be ineffective in preventing fouling. Some experience is necessary to know when a paint has been properly mixed, but it can always be taken for granted that it is not ready for applica-

tion so long as sediment or lumps can be felt at the bottom of the container with the mixing paddle, or so long as the paint looks streaky. No harm can come from more mixing than is necessary, but the durability of the paint film will be much impaired if mixing has not been sufficient.

4. At times small quantities of paint must be made on board ship from pigments and oil. Mixing paints by hand does not give very satisfactory results, especially if dry pigments must be used. It is particularly difficult to thoroughly incorporate dry red-lead, or even paste red-lead, with linseed oil by hand. Such paints should be mixed by mechanical means. For this reason ships should make requisitions for ready-mixed paints, especially red-lead paint, as far as possible. When red-lead paint must be made from paste or dry pigments on board ship, or for that matter any other paint, it should be appreciated that the process cannot be hurried. The pigment should be added to the oil in small quantities at a time. The mixing container must be sufficiently large to prevent slopping when using a quick, jerky motion of the paddle. So long as dry particles appear, and so long as the mixture looks streaky, the pigment has not been uniformly incorporated with the vehicle.

5. *Care of Paints.*—If a container of paint or varnish has been opened and the contents partly used it should be covered and kept as airtight as possible to prevent scum from forming on the surface. This scum forms due to the partial oxidation of the top layer of oil. If scum is formed, or if foreign substances have become mixed with the paint, it should, before being used, be strained through fine-gauge wire cloth or cheese cloth. Unused paint should always be poured back from the pot into the stock container. The pot should be wiped out with a paint brush before being set aside.

6. Shellac should not be unnecessarily exposed to the air, as the alcohol evaporates quickly, leaving the shellac in a thick, stringy condition, unsuitable for application. Shellac should be kept free of nails or any other iron substance, as iron discolors shellac. Shellac or varnish should not be applied on a damp day, and especially should not be applied over unseasoned or damp wood.

7. *Painting.*—A few instructions as to the best practice in applying paint will be of value in supervising the work of enlisted men and others who are not painters by trade.

(*a*) The brush should be held by the handle and not by the stock. If the brush is held by the stock the hands become covered with paint, which may cause poisoning, especially if there are abrasions on the hands.

(*b*) The brush should be held at right angles to the work, with only the ends of the bristles touching the surface. It should be lifted clear of the surface when starting the return stroke. If the brush is held obliquely to the surface, and not lifted, the painted surface will be uneven and dauby in appearance.

(*c*) The brush should never be completely filled with paint. Only the ends of the bristles should be dipped into the paint. The brush should not be dipped again until the preceding charge has been completely exhausted.

(*d*) In applying the priming coat to steel or the anti-corrosive coat to ships' bottoms it is particularly important that the paint should be well brushed out so that no holidays will remain offering an opening for corrosion. On old work, from which scale and rust has been removed, this is of particular importance, as the surface is bound to be uneven, necessitating thorough brushing so as to cover all spots completely.

(*e*) In applying paint to wood, long strokes, parallel to the grain, should be used. Also, in applying the finishing coat to steel surfaces long strokes are necessary if the work is to have a smooth appearance. The work should be laid off so that the brush can be drawn along the whole length of the particular panel being painted. This results in fewer breaks in the lines.

(*f*) The work should be started by laying on the paint with parallel strokes, then crossing this application with parallel strokes at right angles. Medium pressure should be used during the first application and a light pressure when the work is crossed. All final laying on should be in the lengthwise direction of the work.

(*g*) When painting overhead surfaces, ceiling panels should, as far as possible, be laid off fore and aft and the beams athwartship. Where panels contain a great many pipes, running parallel with the beams, it may be difficult to lay off the ceiling panel fore and aft. In such cases better results will be obtained by laying off parallel with the beams.

(*h*) When painting vertical surfaces, bulkheads, etc., the work should be laid off vertically. In all cases each succeeding coat of paint should be laid off in the same direction. The painting should be started at the top of the vertical surface, working downward.

(*i*) The paint in the pot should be kept well mixed while the work is proceeding.

(*j*) Where appearance is the principal consideration, as in the living spaces or on the top side, it should be remembered that a heavy coat of paint will show brush marks and will present an uneven finish. Better results will be obtained by applying two coats of thin or medium paint than one coat of heavy paint.

(*k*) A succeeding coat of paint should never be applied until the previous coat has dried thoroughly. Paint dries from contact with the air. The drying of the first coat will be retarded if the second coat is applied too soon. Any coat of oil paint, free from non-drying oils, will appear perfectly dry in a few days, when as a matter of fact the complete oxidation of linseed oil to linoxyn has not taken place. Whenever possible red-lead should be given three weeks to dry before the second coat is applied. Where red-lead is used for touching up the upper works of ships, this cannot always be done because the surface presents an unsightly appearance, but in any case as much time as possible should be allowed before the spots which have been touched up are given the coat of finishing paint.

(*l*) Waste, rags, etc., covered with paint, varnish or oil, should never be allowed to lie about in paint rooms or other enclosed spaces on board ship or ashore, as they are a serious fire risk. This is because the drying of paint is similar to the burning of wood; both are oxidizing processes—heat being generated in both cases. When a paint film dries the heat radiates readily because the area is large and consequently there is no perceptible rise in temperature. The case of a bunch of waste or of rags, covered with paint or linseed oil, is, however, different. The heat generated in drying process cannot radiate freely, the temperature gradually rises and finally the whole mass bursts into flame. This phenomenon is often referred to as spontaneous combustion.

(*m*) The volatile thinners used in paints are inflammable. Flames should, therefore, be kept away from open containers of paint.

8. *Varnishes.*—The wood which is to be varnished must be perfectly dry, clean and smooth. A filler is first applied. Fillers consist of a vehicle mixed with various pigments, depending on the color which the finished article is to have. The pigments most commonly used as fillers are burned umber, raw umber and raw sienna, although there are numerous others. The filler serves two purposes, to stain the wood to the color desired, it being remembered that varnish is practically a transparent fluid, and to close the pores of the wood. The filler is applied with a brush and after it has set should be rubbed down with excelsior or waste. The surface should then be allowed to dry for twenty-four hours, after which it should be sandpapered with No. 00 sandpaper. Any necessary puttying should now be done, the putty having been brought to the same color as the filler. After the putty has dried the surface should be dusted off and then given a coat of clear shellac. When this coat of shellac is dry the surface should again be sandpapered with No. 00 sandpaper, dusted off, and a second coat of shellac applied. When this coat is dry it should be rubbed down with No. 00 sandpaper, dusted off, and the first coat of varnish applied. For outside exposure a spar varnish must be used. If a second coat of varnish is needed, the first coat, after having been allowed to dry for at least six days, should be rubbed down with No. 0 steel wool, and dusted off before the second coat of spar varnish is applied.

9. Furniture, interior work, etc., which is to be varnished should be filled and shellaced as outlined above. At least two coats of varnish should be applied over the two coats of shellac, allowing at least six days to intervene between the two coats of varnish. For a dull finish the surface should be rubbed with pumice stone, soap and water, then washed off and dried with chamois skin. For a glossy finish the surface should be rubbed with pumice stone and crude oil, then rubbed with pumice stone, cleaned with crude oil (8 parts) and mineral spirits (1 part), and finally finished with cheese cloth or clean waste. In place of crude oil with pumice stone a rubbing oil consisting of one-quarter raw linseed oil and three-quarters gasoline will also give good results.

10. It is of the greatest importance that before any coat of varnish is applied the surface should be entirely free from dust

particles and grit, as these will show through if not removed and will also shorten the life of the coating. The varnish should be applied as smoothly as possible by crossing the work and allowing it to flow. It is particularly important that the varnish should not be applied too thickly. Only the tip of the varnish brush should be dipped into the pot so as to prevent the application of an excessive amount. If the varnish is applied too thickly it not only results in bad appearance of the surface, but it also decreases the effectiveness of the coat, because the upper surface will dry first, forming a skin, which prevents air from having contact with the varnish underneath.

11. It should be remembered that varnish which is suitable for interior work will not give satisfaction for outside work. Spar varnishes (exterior varnishes) contain a higher percentage of oil than interior varnishes. China wood oil is also used in such varnishes in place of linseed oil to give the varnish greater durability and resistance to moisture. Varnishes for interior work have a higher lustre and can be rubbed and polished to greater effect than spar varnish, but they are not so durable or so water resistant as the latter.

12. *Bituminous Compositions.*—Before applying bituminous compositions it is absolutely essential that the surfaces be cleaned down to the bare metal and thoroughly dry. The surfaces must be entirely free from all scale, rust, grease, old paint, dust and any other foreign matter. Grease may be removed by scrubbing with gasoline, but if this is done care should be exercised that open lights are kept away from the vicinity where the gasoline is being used. Artificial ventilation must be provided to free such spaces from gasoline vapors as rapidly as possible. While it is of the highest importance to have the metal surfaces over which any protective coating is applied thoroughly bright and clean, this is particularly important in the case of bituminous compositions, because touching up later on is not practicable, as it is in the case of painted surfaces.

13. The bituminous solution should be applied promptly after the compartment has been cleaned and made ready. It must be well spread out with a brush and care must be taken to insure that no spots are left uncovered, otherwise the enamel will not adhere.

After this priming coat has dried, or at least become tacky, it is covered by a coat of hot bituminous enamel. Complete and long drying of the priming coat of bituminous solution is not necessary as it is in the case of a first coat of oil paint, because no chemical reaction, such as the combination of oxygen with oil, is involved. The solution merely forms a binder between the metal and the enamel. By itself it has little merit in protecting the steel surface against corrosion.

14. The bituminous enamel is heated in a pot. As it has a tendency to boil over the pot should be only partially filled. The pot must be kept well stirred during the heating process otherwise the mineral ingredients in the enamel will settle out. It is important that the enamel should not be allowed to become too hot otherwise it will be brittle after cooling. When the enamel begins to froth it is a sign that it is getting too hot. The enamel should first be applied to overhead, vertical, and inclined surfaces by means of a brush. The thickness over such surfaces should not be less than 1/16 of an inch. In the case of flat horizontal surfaces the enamel should be poured on and spread out rapidly with a long handled brush to a thickness of at least 1/4 of an inch. The enamel should also be poured along the lower edge of bulkhead bounding angles, boiler foundations, and other metal work projecting through the enamel in order to completely coat the corners of angles, butt straps, and rivet heads. As the bituminous solution and the enamel are both black, making it difficult to see where the enamel has been applied, special care and inspection are necessary to insure that the entire surface has been coated with enamel. On account of the fumes which arise during the application of the hot enamel artificial ventilation should be provided for the compartment in which the work is being done.

15. *Portland Cement.*—A cement wash should be made with two-thirds Portland cement and one-third fire clay mixed with fresh water to such consistency that it can be applied with a paint brush. The metal must be clean and free from rust as for other protective coatings. Two coats should always be applied. Portland cement when used in bilges and pockets to provide drainage to limber holes, sumps, etc., should be mixed in the proportions of one part of cement to two parts sand. The mortar in which

deck tiling is laid should consist of one part of cement to one part of sand.

16. Whenever a large pocket or the corners resulting from inclined plating is to be filled, coke should be used to reduce the weight of the mass. A coat of cement mortar about one inch thick should first be applied. The space should then be filled with fine coke over which a thin mortar should be poured to fill the voids. The top should then be finished off with a 1-inch coat of fresh mixed mortar sufficiently stiff to be troweled to a surface.

17. Wherever cement is used on iron or steel, the surface of the metal must be thoroughly cleaned down to the bare metal and care must be taken to clean off all oil, dirt, paint, scale, or other foreign matter. Cement should never be applied to a deck preparatory to tiling until the deck has been satisfactorily tested for watertightness.

18. *Care of Brushes.*—Steps should be taken to tighten the bristles of brushes before they are put in use. Even though a brush may be of good quality and well manufactured, it may, after lying in store for a considerable time, shed its bristles. This shedding of bristles is due to the drying out of the handle causing it to shrink and to loosen somewhat the pressure between the ferrule and the handle. The bristles may be tightened by holding the brush in a vertical position with bristles pointing up and wetting the end of the wooden handle inside the bristles with about a teaspoonful of water, then allowing about one-half an hour for the handle to swell. If time is available it is still better to immerse the brush in water for twenty-four hours to the top of the ferrule. To prevent the bristles in a round brush from spreading it is good practice to bind the heel end of the bristles at the ferrule with cotton line. This makes the brush stiffer and holds the bristles together. As the bristles become shorter due to wear the binding should be removed.

19. No matter how good a brush may be it will be ruined very quickly if not properly treated when not in use. A paint brush after use should be thoroughly cleaned in turpentine or with soap and water. If left in water any great length of time the bristles are liable to twist and to lose their elasticity, although this is better than allowing the brush to dry and to lie about for a long time

without washing out the paint. After cleaning the brush it should be kept in a trough containing a sufficient quantity of raw linseed oil to cover about one-half the length of the bristles. Large brushes should have a small hole bored through the handle near the end for suspending the brush, by means of a wire or string, in a trough of oil. Brushes should never be stowed standing in buckets with the weight of the brush on the bristles. Large brushes so stowed soon lose their shape and become useless.

20. Should a paint brush become quite hard with paint it should be soaked for twenty-four hours in raw linseed oil and then in hot turpentine or mineral spirits. This treatment will generally loosen up the bristles.

21. Varnish brushes should be suspended in the same kind of varnish for which they are used. If this method is not possible, boiled linseed oil will be found reasonably satisfactory. If a varnish brush is thoroughly cleaned with turpentine or gasoline, followed by washing with soap and water, it may be kept in the dry.

22. Lettering brushes should be washed in turpentine or in gasoline until clean. If they are not to be used for sometime they should be dipped in olive oil and smoothed from heel to tip. Such brushes can then be stowed away for future use.

23. Shellac brushes should be kept in a small amount of mixed shellac or alcohol. Such brushes should never be put in water. If the brush is not to be used for some time it should be thoroughly cleaned in alcohol and then stowed away in the dry.

THE DEVELOPMENT OF THE WRIGHT AIR-COOLED AVIATION ENGINE.

CAPT. H. C. DINGER, U. S. N., MEMBER.

At the close of the World War the American aircraft engine industry appeared to be committed to the policy of concentrating on the water-cooled type of engine to the complete exclusion of the air-cooled engine. The only successful air-cooled aircraft engine built in the country up to that time had been of a rotary type, and when it became evident that the rotary type was not suitable for the larger powers required in the later types of aircraft there was no other type of air-cooled engine available to replace the rotary. A survey of conditions and opinions prevailing at that time would certainly have indicated that the future of the air-cooled engine was far from promising.

On the other hand, this opinion was not unanimous, and there were a few engineers in the aircraft engine industry who believed that possibilities of the air-cooled engine should not be overlooked, in spite of the apparent superiority of the water-cooled type of that date. Probably the leader in this school of thought was Mr. Charles L. Lawrance, an engineer of high standing, who had been engaged in the development of experimental air-cooled aircraft engines for many years. In both the Army and Navy technical aircraft organizations there was a general opinion that, while the development of a satisfactory air-cooled engine offered many difficulties, the advantages to be derived should such a development prove successful, were sufficient to justify the undertaking, even though the final outcome was somewhat doubtful. This opinion was undoubtedly supported by the knowledge that in Europe the air-cooled engine was receiving a great deal of attention and showed promise of developing into a very satisfactory type.

Several experimental developments were undertaken, most of which proved to be unsound, but as a result of this work a nine-cylinder radial engine was designed and constructed by Mr. Lawrance on an experimental contract with the Engineering Division

of the Army Air Service. This engine completed a very satisfactory 50-hour endurance test, at a power output, weight and fuel consumption comparable with the best water-cooled engines of that date. This accomplishment represented the first real success in this country in the development of a non-rotary type of air-cooled engine, and it is undoubtedly to the success of this experiment that we owe our present position in regard to air-cooled engine development in this country. The original Lawrance nine-cylinder radial engine, therefore, is responsible for most of the subsequent successful development in the air-cooled field. The extent to which this engine has led the development will be appreciated when it is realized that the first Lawrance nine-cylinder radial engine passed a successful endurance test in 1921, and that it was not until 1925 that any other type of American air-cooled aircraft engine successfully completed this test, and even at the present time there is no other American air-cooled aircraft engine which can be said to have yet been thoroughly service tested in the air. In view of the unique position occupied by this series of engines in respect to the general development of air-cooled aircraft engines in the United States, it may prove interesting to outline the history of this development in some detail.

As far back as 1916 Mr. Lawrance was experimenting with radial air-cooled engines, starting with the two-cylinder opposed motor model "A" of 28 horsepower at 1400 R. P. M., which by gradual changes was developed into the successful type of three-cylinder radial engine known as the model "L" of 60 horsepower at 1800 R. P. M. These experiments in aviation radial engines were considerably encouraged by both the Army and Navy Air Services by the purchase and trial of these engines on experimental planes. By 1919 both the Army and Navy Air Services had a number of model "L-2" engines in use in small planes used for messenger and training purposes. Being encouraged by this success, it was decided to proceed with a more ambitious program, and a contract was entered into with the Engineering Division of the Army Air Service for the construction of a nine-cylinder radial engine known as the "R-1". This engine was constructed and subjected to a 50-hour test by the Engineering Division, with highly satisfactory results, the test being con-

ducted during the month of July, 1921. The engine was a nine-cylinder job, with a bore of 4¼ inches and a stroke of 5¼ inches. The total displacement was 670½ cubic inches. The engine developed 147 horsepower at its rated speed of 1600 R. P. M., and weighed approximately 410 pounds. The fuel consumption of the "R–1" on the 50-hour endurance test averaged .52 pound per horsepower hour, and the oil consumption averaged .12 pound per horsepower hour, which is rather high as compared with the latest "J" type engine, which averages .012 pound per horsepower hour. This engine is illustrated in Figure 1. As previously stated, this represented the first successful endurance test on an American-built radial air-cooled engine of over 100 horsepower and gave a tremendous stimulus to the entire air-cooled engine development program.

Early in 1920 the Army Air Service entered into a contract with the Lawrance Aeronautical Engine Corporation for three model "R–1" engines. Simultaneously with this development, the Navy Department on February 28, 1920, gave the Lawrance Corporation a contract for the construction of five nine-cylinder radial air-cooled engines to develop 200 horsepower at 1800 R. P. M. They were similar to the "R–1" except cylinder and stroke were increased to 4½ inches by 5½ inches. These five engines were designated as model "J–1". (See Figure 2.) The first "J–1" engine was delivered in May, 1921, to the Aeronautical Engine Laboratory at Anacostia, D. C., where the engine was subjected to a fifty-hour endurance test. This test was started on December 19, 1921, and completed January 10, 1922. The maximum horsepower developed by this engine was 233 at 1800 R. P. M., the fuel consumption being .53 pound per horsepower hour, and the oil consumption .05 pound per horsepower hour. Even before the completion of the fifty-hour tests on either the "R–1" or "J–1" engine, the development of the radial engine appeared so promising to the Navy Department that on June 30, 1921, they gave the Lawrance Corporation a production contract for fifty model "J–1" engines.

This engine was destined to supersede all other American-built engines in its range of power output, and until the end of 1925 represented the only American-built air-cooled radial type to suc-

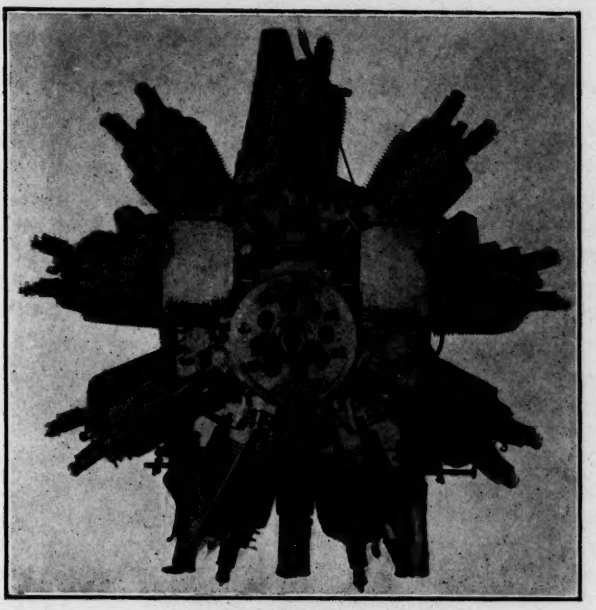


FIGURE 1.
LAWRANCE MODEL "R-1" ENGINE.

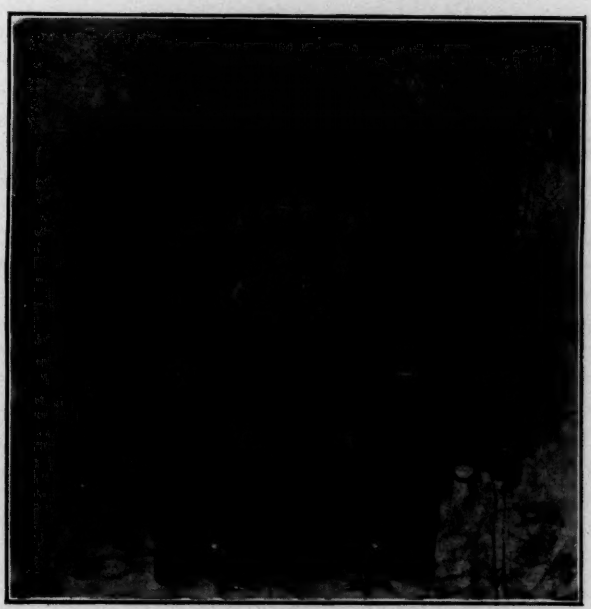


FIGURE 2.
LAWRANCE MODEL "J-1" ENGINE.

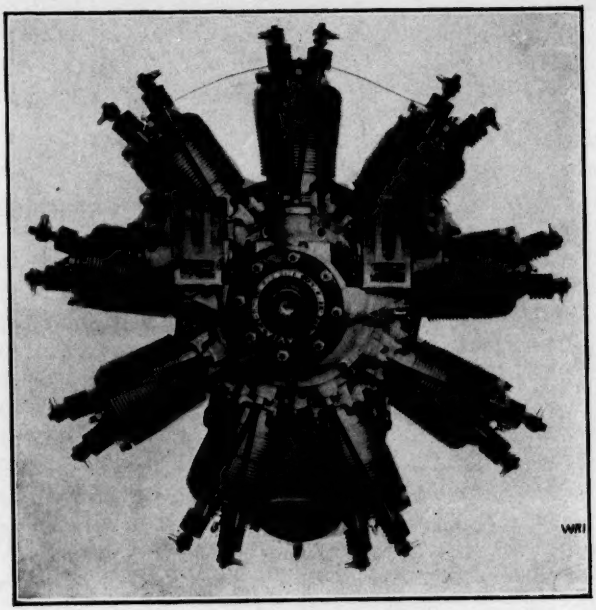


FIGURE 3.
WRIGHT MODEL "J-3" ENGINE.

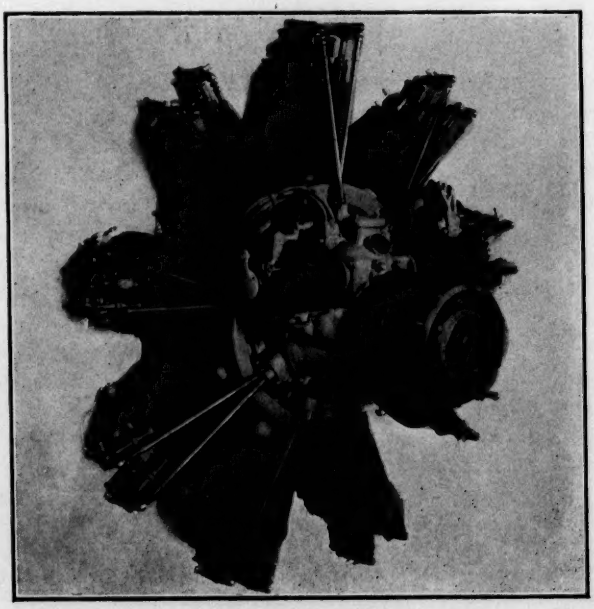


FIGURE 4.
WRIGHT MODEL "J-4" ENGINE.

cessfully complete a standard fifty-hour endurance test. The success of this engine hinged largely upon the cylinder construction, which consisted of a two-valve cast aluminum cylinder, carrying integral fins and provided with a shrunk-in steel liner and cast-in bronze valve seats. The valve gear was of the open push-rod type, and mercury-cooled valves were originally employed. The nine cylinders were equally spaced about a single-throw crankshaft, and the conecting rods consisted of a single master rod with eight articulated rods. The crankshaft was of one piece construction, and the master connecting rod was of the cap type. Three Stromberg carburetors were employed, supplying three induction rings, each of which communicated to three of the nine cylinders. Ignition was supplied by two Splitdorf nine-cylinder magnetos mounted on the nose of the engine, with their axes at right angles to the axis of the crankshaft. Fifty of these engines were constructed for the Navy Department, and, while not entirely free from difficulties, gave such an excellent account of themselves that thoughts of reverting to the water-cooled engine for this power were abandoned.

One of the first production engines on this contract, chosen at random, was subjected to an endurance test which, it is believed, has not been equalled by any endurance test run since on any radial air-cooled engine in this country. This test was to be a three hundred-hour endurance run, divided into three periods of one hundred hours each. The one hundred-hour periods were sub-divided into eighty-six hours of running at 8/10 throttle opening, the engine turning over 1700 R. P. M. and producing 160 horsepower. This corresponded to the power which the engine would deliver at 1700 R. P. M. at 2500 feet altitude when fitted with a propeller which permitted the engine to deliver 200 horsepower at 1800 R. P. M. at sea level. The other fourteen hours were run at full throttle with a propeller permitting the engine to run at 1650 R. P. M. This test was designed to determine the engineering and endurance characteristics of the model "J-1" engine under conditions approximating actual conditions imposed in flight. The test was started on October 25, 1922, and the first one hundred hours of running were finished on October 30, 1922. The average performance during this run was as follows:—

86 hours—165 horsepower at 1720 R. P. M., oil consumption .025 pound per horsepower hour; fuel consumption was .49 pound per horsepower hour.

14 hours—212 horsepower at 1625 R. P. M., oil consumption .015 pound per horsepower hour; fuel consumption .505 pound per horsepower hour. The average brake mean effective pressure during the full throttle run was 132, which compares very favorably with the latest type of "J" engine. Considering the stage of air-cooled engine development at that time, it was really a remarkable performance.

The engine was disassembled for inspection after the one hundred-hour period and was found in very good condition. The only weakness disclosed during this test was in the valve mechanism. Five rocker arms were broken during the hundred-hour run.

The second hundred-hour period was started on November 14, 1922, and sixty-four hours of this run were completed at an average performance of 166 horsepower at 1725 R. P. M.; the oil consumption being .020 pound per horsepower hour, and the fuel consumption .510 pound per horsepower hour. The engine was wrecked on November 17, 1922, in the sixty-fourth hour of the duration test, due to the breaking of the exhaust valve in Number 1 cylinder, believed to have been due to excessive clearance in the valve guide. The valve head broke through the piston head and traveled into the crankcase, thus wrecking the engine.

This engine had run two hundred and one hours all told under its own power before it was wrecked, and examination of the major parts of the engine after the wreck showed them to be in very good condition. Upon the basis of the excellent performance of the "J–1" engine on this test, it was proposed to run a second three hundred-hour endurance test on another production "J–1" engine, boosting the rating of the engine from 200 horsepower at 1800 R. P. M. to 225 horsepower at 1800 R. P. M., the engine to be run at full throttle for the entire three hundred-hour period.

On May 15, 1923, the new three hundred-hour endurance test was started, but after seventy hours the engine developed trouble and further running on it was abandoned. The main trouble was the failure of both of the crankshaft ball bearings, and also the

failure of the cam drive and magneto drive mechanism. The average results during this test were as follows:— 226 horsepower at 1810 R. P. M., fuel consumption .51 pound per horsepower hour, and oil consumption .026 pound per horsepower hour. This test showed that the engine would not stand up under full throttle conditions for any considerable length of time at the increased rating. It was decided, however, to make still one more attempt at a three hundred-hour endurance test with a rating of 225 horsepower at 1800 R. P. M., but this test met with no better success than the previous one, the engine failed this time by the cracking up of the crankcase after fifty-eight hours of endurance running. The following average performance was obtained:— 225 horsepower at 1750 R. P. M., fuel consumption .513 pound per horsepower hour, and oil consumption .021 pound per horsepower hour. These tests showed rather definitely the limit of power at which this particular design could be operated. It also showed that at rated load, as designed, it was quite satisfactory.

In the meantime the Lawrance Corporation was getting out its production engines with very little trouble. In addition to the original contract for fifty engines, the Bureau of Aeronautics had awarded a contract for twenty-five more "J–1" engines on 29 May, 1922, and a further contract for thirty-five "J–1" engines on 19 February, 1923. About the same time the Army Air Service made a contract for ten "J–1" engines, so that a total of one hundred and twenty "J–1" engines were built by the Lawrance Corporation. These engines were all built by the DeLaVergne Machine Company under the supervision of the Lawrance Corporation. Only very minor changes were made in the model "J–1" engine during production, the most important one being the strengthening of the valve-actuating mechanism.

The average performance of the "J–1" engine on the test stand is as follows:—

228 horsepower at 1800 R. P. M., oil consumption .025 pound per horsepower hour, and fuel consumption .52 pound per horsepower hour. Dry weight of engine with ordinary accessories, four hundred and fifty-five pounds.

On January 19, 1923, the Lawrance Corporation received a contract for the production of two model "J–2" engines to develop

250 horsepower at 1800 R. P. M. These two engines were built from standard "J–1" engine parts, the increased horsepower being secured by enlarging the cylinder bore from 4½ inches to 4⅞ inches diameter. The design of the cylinder itself was radically different from the "J–1" design, and was very much similar to the present "J–5" design in that it consisted of a steel cylinder barrel with integral fins and holding down flange and an aluminum head screwed on to the steel barrel. The pistons were of the ribbed type, similar to those now being used on the "J–5" engine. The "J–1" crankcase was used, the cylinder pads being bored out to take the larger diameter cylinder. Otherwise all standard "J–1" parts were used.

These two engines were each given a regular acceptance test and showed the following results: 252 horsepower at 1750 R. P. M., oil consumption being .039 pound per horsepower hour, and fuel consumption .510 pound per horsepower hour. They weighed dry without accessories four hundred and eighty pounds each. They were shipped to the Aeronautical Engine Laboratory at Anacostia, D. C., for further testing. Another interesting experimental project carried on by the Lawrance Corporation during 1923 for the Army Air Service was the development of the model "T" engine, a three-cylinder radial of the same general dimensions as the model "L" engine, the main difference being that in the "T" model the cylinder did not have a steel sleeve, but the piston ran in the aluminum bore. The bearing surface in the bore of the cylinder was somewhat hardened by rolling the metal to a depth of about .005 inch. Three of these engines were built and passed satisfactory acceptance tests and were then forwarded to McCook Field for further testing. These engines were designed for use with radio-controlled torpedo planes, and for that reason were much cheapened as to construction.

It might be interesting to mention here that early in 1923 the Lawrance Corporation, at the request of the Navy Department, designed a five-cylinder radial engine to produce 60 horsepower at 1800 R. P. M., to replace the model "L" engine. This design was submitted to the Bureau, but the Lawrance Corporation was not awarded the contract because Mr. Noble of the Kinney Manufacturing Company had submitted at the same time a design

of an engine of the same horsepower and incorporating an Oleo valve gear, giving a much better horsepower-weight ratio. The Kinney Manufacturing Company built six of these engines, and it is understood that they are giving satisfactory service.

About the middle of 1923 the interests of the Lawrance Corporation and the Wright Aeronautical Corporation were merged, but the Lawrance Corporation completed all its contracts for "J–1" engines, amounting to one hundred and twenty engines, before its existence ceased. Immediately after the merger of the two companies the Navy Department awarded the Wright Corporation a contract for fifty model "J–1" engines. The Engineering Department of the Wright Corporation so changed the design of this gine that it was necessary to redesignate the engine model "J–3." The most important change was the strengthening of the crankshaft by increasing the main bearings ¼ inch in diameter. This necessitated the re-arrangement of the cam and magneto drive mechanism. The engine is illustrated in Figure 3. The cylinders were improved by providing a bronze spark plug bushing, a harder bronze for the valve seats and thickening the metal around the combustion chamber and valve seats. The general accessory layout was simplified and improved, and the Wright patent timing clutch was provided in order to simplify the adjustment of the valve timing. A Stromberg NA–U5 carburetor was substituted for the three single carburetors used on the original model. The lubrication system was altered considerably by relocating the pumps and strainers, and improving the method of pressure regulation. Fifty of these engines were manufactured.

The average maximum performance of the "J–3" engine on test is as follows:—211 horsepower at 1800 R. P. M., fuel consumption .525 pound per horsepower hour, and oil consumption .022 pound per horsepower hour. Some trouble was encountered with the "J–3" engine during production in that the oil consumption ran high during the summer months.

The next step in the development of the "J" series was the model "J–4". Here again all of the essential features of the design were retained, but numerous minor improvements were incorporated as a result of service experience with the preceding model. The cylinder design was modified by substituting the screwed-in

barrel for the shrunk-in barrel, and the hold-down flange was made integral with the barrel instead of forming a part of the aluminum head. The valve gear was improved by increasing the area of the rubbing surfaces, in order to reduce wear. An improved piston design, using a stronger and harder alloy, was developed. Minor changes were made in the crankcase to improve accessibility and ease of adjustment. An enamel finish was added to all exposed parts. This engine is illustrated in Figure 4. One hundred and ninety engines of this model were manufactured and their performance in service has been extremely gratifying.

In eliminating the hold-down flange on the aluminum cylinder barrel, the pattern was slightly modified so as to make it easier to cast. This eliminated some of the cooling fin area near the exhaust port and caused considerable trouble in the production of these engines during the hot weather months in the over-heating of the cylinder heads. This necessitated an increase in fuel consumption from .55 pound per horsepower hour to .65 pound per horsepower hour, which was granted by the Bureau of Aeronautics, after it was found that these engines would not run successfully on the lower fuel consumption in hot weather.

Another difficulty encountered during production on the "J–4" model was the failure of several engines due to insufficient lubrication at the knuckle pins. This was finally remedied by providing pressure lubrication to the knuckle pins in place of the splash lubrication previously provided. This was accomplished by cutting a groove in the back of the master rod bearing, opposite the metering oil hole in the crankpin, and drilling a hole through the bearing. From this groove the oil was led by means of drilled holes to each individual knuckle pin. This method of lubricating the knuckle pins and bushings has proved very successful, and these parts have not given any trouble since the change was introduced.

In point of performance, the "J–4" model was similar to the "J–3". The average maximum performance on test was as follows:—215 horsepower at 1800 R. P. M., fuel consumption, first one hundred and ten engines, about .535 pound per horsepower hour, the remaining engines (forty-five) (built during the hot summer months) .60 pound per horsepower hour. The average oil consumption was about .012 pound per horsepower hour.

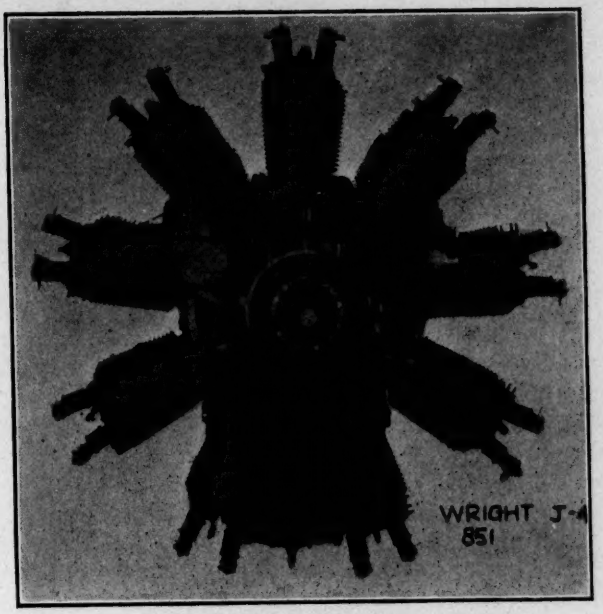


FIGURE 5.
WRIGHT MODEL "J-4A" ENGINE.

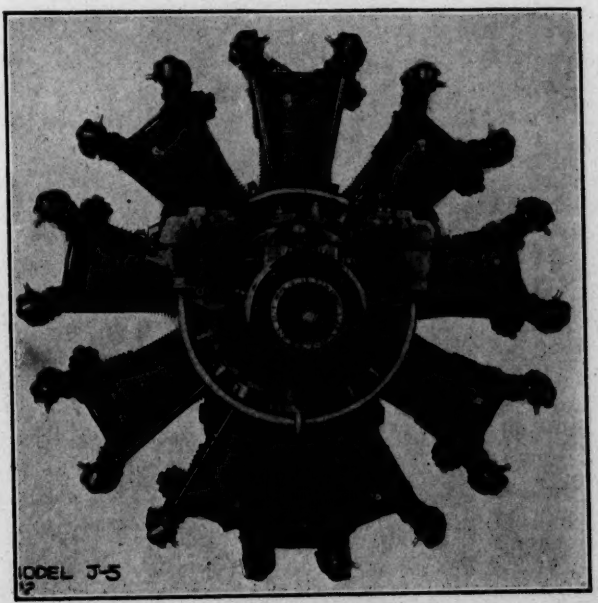


FIGURE 6.
WRIGHT MODEL "J-5" ENGINE.

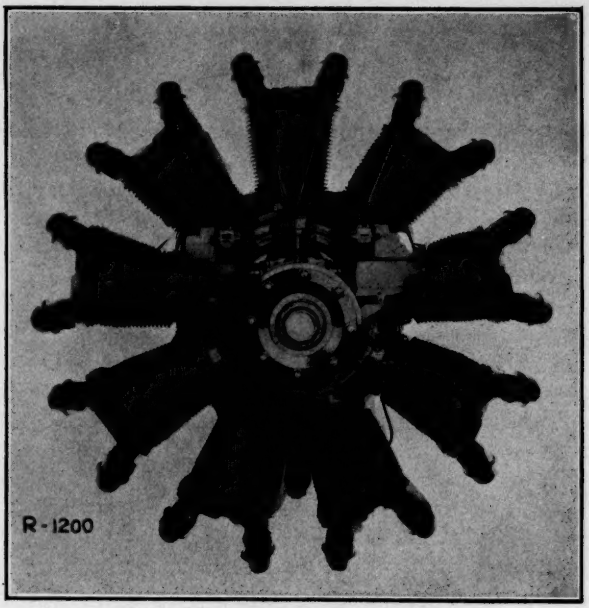


FIGURE 7.
WRIGHT MODEL "R-1200" ENGINE.

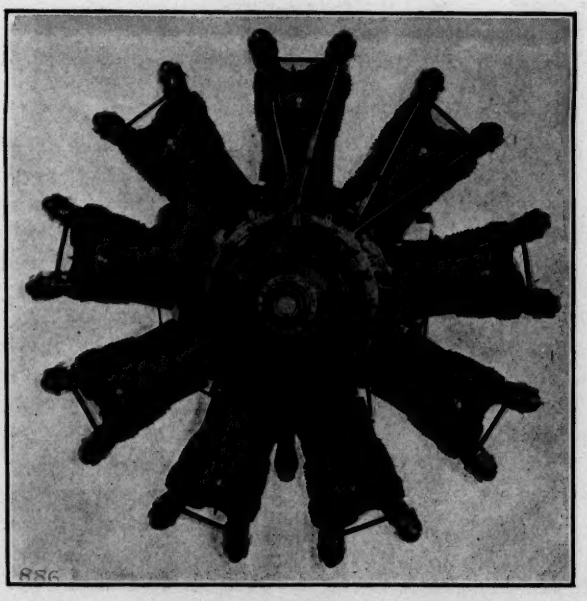


FIGURE 8.
WRIGHT MODEL "P-2" ENGINE.

The successor to the "J–4" was the "J–4A". The changes incorporated in the latter model are of a very minor nature, such as the substitution of duralumin for steel in the front propeller hub plate, the addition of cowling supports to the crankcase, provision for fuel pump drive, the substitution of Scintilla magnetos for the Splitdorf magnetos and a change in the magneto coupling. Practically the only change aiming at greater strength and durability was in the substitution of a wrist pin of larger diameter. This engine is illustrated in Figure 5. One hundred and seventy-five of these engines were constructed and are being placed in service.

The important difference between the "J–4" and the "J–4A" model was in the aluminum cylinder barrel. In the "J–4" model the cooling fins were spaced close together, the lower fins being finish-machined. In the "J–4A" model the fins were spaced wider apart and cast thin so as to eliminate the machining operation. While this type of cylinder was thought to be cheaper to produce, there was considerable trouble in production. It was necessary to cast the aluminum alloy several hundred degrees hotter in order to get a good form of fin. This had a tendency to make the head of the casting porous and leaky under test. In very hot weather it was found that the head could not be kept sufficiently cool except by increasing the fuel consumption considerably. It is believed, however, that this is a sea level difficulty and that in flight the engines will run cool even in the tropics. This is borne out by service performance. In an attempt to remedy these conditions, the cylinder was redesigned, the number of fins being increased to the same number as the "J–3" model, also additional fins were placed around the inlet and exhaust valve parts. This cylinder was used very successfully on "J–4A" commercial engines. This cylinder design is known as "J–4B".

An endurance test run on a "J–4A" engine with "J–4B" cylinders was run with fuel consumption at .50 pound and no overheating was observed. Considerable trouble was encountered in production on this type cylinder by reason of leaks past the thread where the sleeve is shrunk in. This was finally overcome, and the last few Navy "J–4A" engines and all replace cylinders will be this "J–4B" type. These difficulties are mentioned to show that this development was not all plain sailing and that numerous obsta-

cles and production difficulties had to be met, and as the Inspector on the contracts in question I can state that they were all finally overcome.

The development of the "J" series up to this point had served to convince the American aeronautical profession that the air-cooled engine was not only a feasible type, but that it possessed many inherent advantages over the water-cooled engine. There were, however, two respects in which these engines did not compare favorably with the water-cooled variety. It is evident that an open valve gear exposed to moisture, dirt, etc., cannot possibly be as durable as a completely enclosed gear, such as is employed on the water-cooled engines, and the "J–4A's" do not show quite as low a fuel consumption as may be obtained from the best water-cooled engine of the same power output. In an effort to remove these two remaining handicaps, and at the same time still further increase the endurance of the "J" series, the "J–5" model was developed. This engine consists essentially of a "J–4A" crank-case, equipped with an entirely new variety of cylinder and valve gear. The cylinder construction consists of a steel barrel with integral fins, provided with a cast aluminum head screwed and shrunk to the upper end of the barrel. The total cooling surface is greatly increased over that provided on the "J–4A," and the port arrangement has been modified to further improve the cooling of the head. Salt-cooled valves are utilized, which result in maintaining the exhaust valve temperature below red heat at all times. The valve gear is entirely enclosed, and all wearing surfaces throughout the gear have been generously increased. The piston design has been modified, and a superior alloy employed with a view to reducing the working temperature and improving the wearing qualities of the piston. This model has also been provided with a new type of carburetor developed by the Stromberg Company, which incorporates in a single unit three barrels supplied by a common float chamber, each barrel communicating to three cylinders through a separate manifold. This arrangement results in very satisfactory distribution and avoids the complication involved in using three separate single-barrelled carburetors. This engine is illustrated in Figure 6. While the "J–5" engine is still in an experimental stage, indications are that the features aimed at in

its design have been fully realized. The fuel consumption is quite on a par with the best water-cooled engines, and the enclosed valve gear has proven extremely durable. This model can be compared with the best water-cooled engines of the same horsepower from any point of view, and will be found to at least equal the performance of the water-cooled type, and at the same time all of the advantages inherent in the air-cooled design are retained.

Two fifty-hour tests have so far been run on the "J–5" engine. The first endurance test was run at 2000 R. P. M. at full throttle and gave an average mean effective pressure of 121 pounds per square inch. The second endurance test was run at 1800 R. P. M. at full throttle in an attempt to increase the mean effective pressure. One slightly lower was obtained, however, the average being 119 pounds per square inch. This was because of the low fuel consumption, which had a tendency to overheat the cylinders and hence the drop in mean effective pressure.

The remarkable performance on the second endurance test, however, was that the entire test was run at a fuel consumption averaging .45 pound per horsepower hour, and the cylinder temperature was not above 500 degrees F. in the head. In this respect the "J–5" engine is a marked improvement over the "J–4" and "J–4A" types.

In the second endurance test of the "J–5" engine an Allison steel-backed "kelmet" lined master rod bearing was used, and after the fifty-hour test this bearing was in excellent condition. The babbitt-lined bearings heretofore used were always more or less damaged after an endurance test, the babbitt cracking up. Some difficulty has been experiencd in obtaining sound piston castings of "Y" alloy. "J–5" engines are now in production. The "J–5" is an improvment over the "J–4A," in giving a higher horsepower, less fuel consumption, a more reliable valve operating gear and a greater endurance due to improved cylinder design.

GENERAL COMMENTS ON "J–4" AND "J 4A" WHIRLWIND ENGINES.

A total of four hundred and fifty-seven of this type of engine have been built at the Wright plant for the following customers:—

295—For U. S. Navy, Bureau of Aeronautics.
53—Stout Engineering Corporation (Ford).
30—Fokker Company.
14—Huff, Daland Company.
10—Argentine Government.
8—Canadian Government.
7—Peruvian Government.
4—Byrd Arctic Expedition.
4—Wilkins Arctic Expedition.
4—Cuban Government.
28—Miscellaneous commercial enterprises—mostly for air mail carriers.

This will indicate that this type of engine has been on a good production basis and that a sufficient quantity has been built and operated to give an opportunity to develop all weak points by actual service test. This design of engine may, therefore, be considered as a fully developed type and one having excellent reliability. It has been flown all over the world, including the North Pole.

The "J-1," "J-3" and "J-4" model engines had bronze-backed babbitt-lined master rod bearings. These did not prove entirely satisfactory on duration running and in service, the bronze not being stiff enough to hold the bearing from going out of shape slightly. In the middle of the "J-4" contract steel-backed bearings were tried and found better than the bronze-backed bearing, though on endurance test the babbitt was still found slightly cracked, but not as badly as was formerly the case when bronze-backed bearings were used. About eighty of the "J-4" and all the "J-4A" engines were equipped with the steel-backed babbitt-lined bearings. The "J-5" engines also have the steel-backed babbitt-lined bearings. On the second endurance test run on this engine a steel-backed kelmet lined bearing was tried and at the end of the run the bearing was in excellent condition.

LARGER AIR-COOLED ENGINES.

"J-4" and "J-5" engines may be considered an entirely successful type of air-cooled engine for two hundred horsepower. It is, however, desirable to obtain engines of higher power so that

the benefits of air-cooled power plants may be secured on larger and faster planes. This has resulted in the Wright Company, in conjunction with the Bureau of Aeronautics, developing several larger air-cooled engines which will now be described briefly.

"R–1200".

This is a twelve hundred cubic inch piston displacement engine rated at 350 horsepower at 1900 R. P. M., 5½-inch bore by 5½-inch stroke, 5.4 to 1 compression ratio, similar in design to "J–5" engine with the addition of a Moss type General Electric supercharger and illustrated in Figure 7.

One "R–1200" engine has been built and tested both on test stand fifty-hour test and flown in the Wright Apache pursuit plane with very successful results. The engine is being placed in production to a limited extent in a Navy contract.

"P–1" AND "P–2" ENGINES.

In these developments the basic idea of the original "J–4" engine has been maintained, nine cylinders acting through a master connecting rod and eight articulating rods. The "P–1" engine was built, tested and successfully flown in a DT plane, and one is now being installed in a Boeing plane.

The "P–1" design may be considered as quite successful and can be applied to planes previously using Liberty engine where it should give a better performance due to less weight, greater ease of installation and the absence of water-cooling complications.

"P–1" ENGINES.

This is a sixteen hundred and fifty cubic inch piston displacement engine rated at 400 horsepower at 1650 R. P. M., 6-inch bore by 6½-inch stroke, 5.4 to 1 compression ratio. In this engine all the accessories including the magnetos are carried in the rear of the main crankcase which is so designed as to include all accessory drives. This makes for a rather complicated crankcase design. The cylinder design consists of a steel barrel with integral fins on which is screwed an aluminum head. Magnesium was tried in the first engine design on the crankcase rear section, inter-

mediate section and front section and a few of the minor castings. The intermediate section broke during the endurance test and was replaced by the regular aluminum alloy. In the subsequent "P–1" engine magnesium was eliminated also from the front and rear sections on account of the difficulty of securing sound castings. In the " P–2" engine no magnesium at all was used.

Average results of fifty-hour test of " P–1" engine are as follows:—

At 9/10s load 363 horsepower at 1650 R. P. M., oil consumption .0417 pound per horsepower hour and fuel consumption .475 pound per horsepower hour.

At rated load, 405 horsepower at 1650 R. P. M., oil consumption .0423 pound per horsepower hour and fuel consumption .546 pound per horsepower hour.

At full throttle 406 horsepower at 1658 R. P. M., oil consumption .0421 pound per horsepower hour, and fuel consumption .594 pound per horsepower hour. Brake mean effective pressure—117.3 pound per square inch.

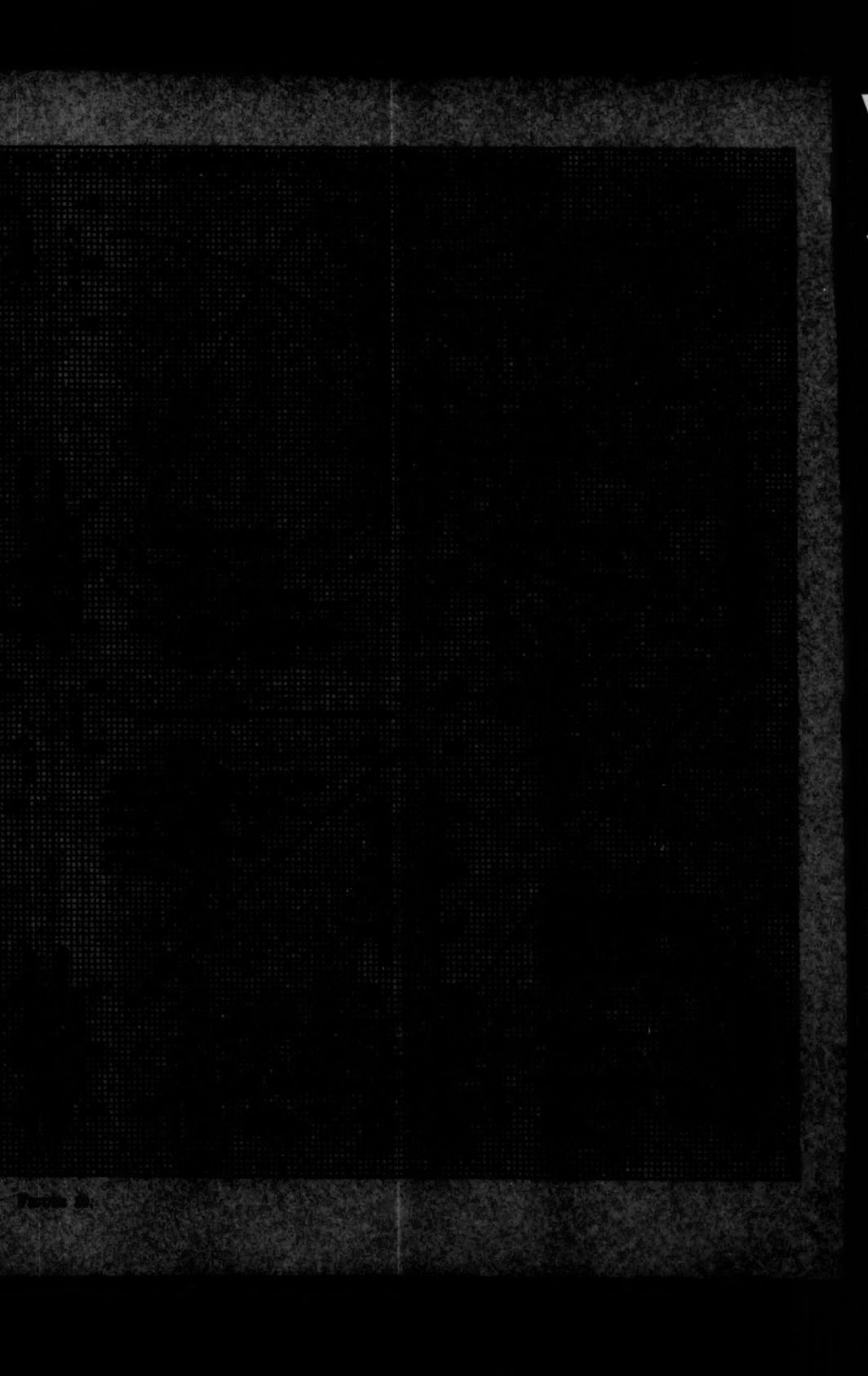
" P–2" ENGINES.

The " P–2" engine is similar to the " P–1" in design. The main differences are the addition of a Moss type supercharger and an enclosed valve gear similar to the " R–1200" design. The intermediate section is not bolted directly to the main crankcase, but by means of long studs is held between the front section and the main case. The cylinder construction is the same as in the " P–1" engine.

The development of this engine is still in process and has not yet been completed (June, 1926). Tests have so far indicated that a stronger master rod than that originally designed is required. This engine is illustrated in Figure 8. Two geared " P–2" engines are under construction, the reduction gear ratio being 2 to 1. A planetary reduction gearing of the Aeromarine type is being used.

COMPARATIVE PERFORMANCES.

The performance curves of the various engines mentioned are shown in Figure 10.



DEVELOPMENT OF CYLINDER DESIGN.

The illustrations in Figure 9 give a cross section of the successive cylinder design used in this development of radial air-cooled engines.

The use of steel and aluminum in varying ways to secure the best cylinder on the least weight is well shown here. The " J–5" type, also used in the " R–1200," has given the best performance.

The successful use of cylinders of larger diameter than those used on " P–1" and " P–2" engines is a matter to be demonstrated by future experiments. It is not believed that the limiting diameter has yet been reached, although perhaps it is being approached. The steel cylinder barrel with fins machined from the forging makes it possible to secure more cooling surface than when the fins are cast on an aluminum barrel. It also does away with the double thickness of metal. It may also develop that a better cooled head would be secured if the fins on the aluminum head were machined out of the casting. Improvement in casting also might enable these fins to be cast thinner and closer together. As far as cooling is concerned, sea level conditions are, of course, more severe than actual flight. It will be noted that this development of cylinder design has been largely by actual tryouts on many different engines. The desired result also has often been thwarted by reason of production difficulties.

EFFECT OF AIR-COOLED ENGINE DEVELOPMENT.

The success of the " J" type of engine changed the entire attitude toward air-cooled development and encouraged other efforts along these lines. As one step after another proved successful, the general attitude of the industry gradually changed from the extreme pessimism of the early days. The prevailing state of mind during the period from the close of the war up to date might be divided into four fairly distinct phases. First, an attitude of pessimism when it was unusual to encounter any opinion favorable to the air-cooled engine. This situation continued up to the first successful fifty-hour test of the Lawrance engine. During the second phase, the prevailing opinion gradually changed until the possibilities of the air-cooled engine in the smaller sizes and

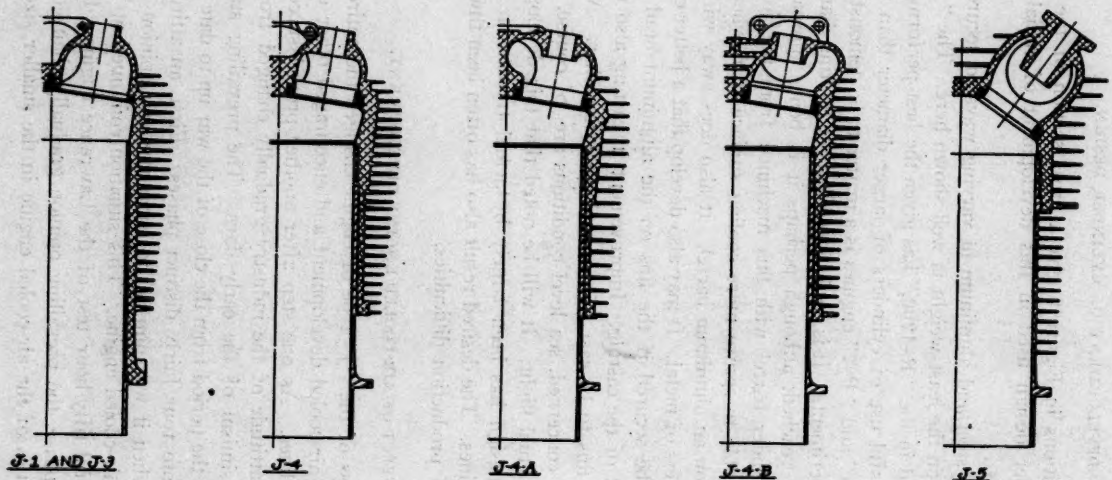


FIGURE 9.

CYLINDER SECTIONS OF WRIGHT "J" SERIES ENGINES, SHOWING DEVELOPMENT OF CYLINDER DESIGN AS FINALLY REPRESENTED IN "J-5" CYLINDER.

for certain classes of service were generally recognized. During the third phase, the conviction gradually pervaded the aeronautical industry that the air-cooled engine was perfectly feasible in any size up to 500 horsepower, and that in many respects the air-cooled type was to be preferred. The fourth and final phase dates from the early part of 1925, and is characterized by the general opinion that the air-cooled engine may eventually displace the water-cooled engine in nearly all sizes and classes of service.

The air-cooled engine offers so many advantages over the water-cooled type that it is difficult to arrange the various favorable points in anything approaching their order of importance, and there is no doubt that the relative importance of the various features will vary with the type of service contemplated. Perhaps, the outstanding advantage of the air-cooled engine lies in its superior reliability. Long endurance runs have demonstrated that the engine itself is quite on a par with the best water-cooled engine with regard to the reliability of its component parts. In addition to this feature, we have complete freedom from operating failures due to the water-cooling systems. The engine is much less sensitive to changes in climatic conditions, as there is no trouble from freezing or boiling of the cooling medium. The air-cooled power plant is much less vulnerable to gun fire than is the water-cooled. A bullet striking either the radiator or engine on the water-cooled power plant is almost certain to cause a forced landing, whereas with the air-cooled engine the bullet must strike the engine proper in order to interfere with its functioning. The cooling system, consisting of metal fins is not apt to be damaged by rifle fire. The shooting away of several fins would have no appreciable effect on the operation of the engine, whereas the slightest leak in the water-cooling system would result in a complete failure.

From the point of view of total power plant weight, the air-cooled engine is greatly superior. The present indications are that the total weight chargeable to power plant may be reduced by at least twenty-five per cent over the best water-cooled installations available. An additional point of advantage appears to lie in the fact that, apparently, air-cooled engines may be successfully operated without the use of oil coolers, an item which is now considered essential in high duty water-cooled power plants. Another

advantage, from the operating point of view, lies in the fact that the air-cooled engine may be warmed up much more quickly than can the water-cooled. This is particularly noticeable when operating in cold climates.

Indications are that the air-cooled engine in the radial form will retain its power at altitude better than the water-cooled engine. This is probably due to the fact that the friction horsepower is considerably lower than that of the water-cooled engine of equal power. This is not surprising when we consider that in the twelve cylinder water-cooled Vee we have seven main bearings, a thrust bearing, six connecting rod bearings and twelve or fourteen camshaft bearings, whereas the nine-cylinder radial air-cooled engine has one main connecting rod bearing, two main crankshaft bearings, one thrust bearing and one cam bearing.

Another advantage with the air-cooled radial lies in its extreme shortness, which, together with a considerable reduction in weight, permits a very compact and maneuverable airplane for a given carrying capacity. This feature also effects a considerable saving in the space required for housing or storing the airplane, and, in general, facilitates handling and transportation.

The advantages of the air-cooled engine are not confined to its behavior in the air, but we find that on the ground the maintenance difficulties are greatly reduced; first, by the elimination of the cooling system, which is essentially a fragile structure, and second, the engine itself is generally more accessible and easier to maintain, especially as regards the accessibility of carburetors, spark plugs, valve gear adjustment, etc. The use of individual cylinders and push rod valve gear also materially assists in reducing the cost of engine maintenance. As mentioned above, the air-cooled power plant usually results in a smaller airplane to accomplish the same purpose, which further tends to decrease the maintenance difficulties and cost of operation.

Considering the matter from the point of view of ease of manufacture and cost of production, the air-cooled engine again offers distinct advantages. The chief superiority in this respect lies in the cylinder construction, which is invariably of the individual cylinder type. The castings are small, light and simple, and when compared with the large, highly complicated castings employed in

the composite block of the water-cooled engine, their advantages from the point of foundry production are obvious. Moreover, a small error in machining or the uncovering of an unsuspected defect as the machining progresses, is not such a serious matter when only a single head casting has to be scrapped, as compared to scrapping a block casting for the water-cooled engine. The use of individual cylinders also simplifies the equipment necessary for manufacturing on a production basis, and considerably facilitates the handling of the work in the factory. In addition to the advantages derived from the use of individual cylinders, the air-cooled engine possesses many other points of superiority from the manufacturing point of view. The crankcase castings are much smaller, the crankshaft is greatly simplified, and in general the absence of large complicated units is noticeable throughout the design. This results in a further reduction in the cost of tools and fixtures, and simplification of the problem of handling the work in the factory. Taking all these factors into account and remembering that the air-cooled engine entirely eliminates the cost of radiator and water piping, it is apparent on the basis of equal horsepower and equal production, the air-cooled engine will show a substantial saving in first cost. When the additional saving due to the reduction in airplane size is added to the saving in power plant cost, the overall economy of the air-cooled engine becomes apparent.

Coming now to the matter of overhaul, as distinct from maintenance, of the engine in a plane, the air-cooled engine again scores a considerable advantage. The fact that large cumbersome units are absent from the structure facilitates the handling and cleaning of parts. Painstaking and laborious fitting of connecting rod bearings and main bearings is reduced to fitting one master rod bearing and merely checking the roller bearings for wear. In overhauling the water-cooled engines with the cylinders *en bloc*, it frequently happens that a defect exists, which necessitates either replacing a block assembly or laboriously disassembling the block and refitting the salvaged parts into a new block. With the individual cylinder construction employed in air-cooled engines, this difficulty is completely avoided, and the cost of spare parts is thereby considerably reduced.

In the light of the above facts, some of which have long been recognized and some of which are only beginning to be appreciated, it should now be conceded that the air-cooled engine is lighter, more reliable, more adaptable to various variations in climatic conditions, cheaper, easier to maintain, easier to overhaul, and for a given load results in a lighter, more compact and handier airplane.

The numerous advantages of the air-cooled engine are now rapidly being recognized by those interested in aircraft engine development, and at the same time we are finding that many of our early opinions regarding the disadvantages of this type were entirely unfounded. The chief objections which were formerly raised against air-cooled engines were based on the idea that this type offered greater parasite resistance and inferior visibility. Both of these criticisms have proven to be unsound as the development of the engine has progressed. Greater compactness has been attained, and each new installation tested adds to our confidence that early fears were unfounded. In practically every case, in which a direct comparison has been possible, the air-cooled engine has equalled the best water-cooled installations as regards speed, and has excelled them as regards ceiling and rate of climb. The extremely short fore and aft dimensions of the radial engine permit considerable rearrangement of the fuselage layout, with the result that instead of finding the visibility inferior, it is generally actually superior to that obtained with the water-cooled engine. Another point formerly raised against the air-cooled engine was the claim that this type of engine would not permit the use of high altitude superchargers. Recent tests, both in this country and abroad, indicate that this contention was unsound, and that we need have no further fears on this ground.

The present status of air-cooled engine development in the United States may be briefly summed up as follows: As previously stated, the first successful fifty-hour test on an American air-cooled aircraft engine was completed by a Lawrance engine in 1921. A period of four years elapsed before this success was duplicated by any other design of American manufacture. In 1925, a small radial engine, designed by Warren Noble and constructed by the Kinney Manufacturing Company under a develop-

ment contract with the Bureau of Aeronautics, completed a very successful test. This was promptly followed by a successful endurance test on the air-cooled Liberty engine, a modification of the standard Liberty engine brought out by the Engineering Division of the Army Air Service. A few months later, a 350 horsepower nine cylinder radial, manufactured by the Wright Aeronautical Corporation, and known as the model "R–1200", successfully passed a standard endurance test. (Figure 7.) This was followed within a few weeks by a successful test of a new engine produced by the Pratt and Whitney Aircraft Company (Wasp). This was also a nine cylinder radial, designed and developed by Mr. George Mead, with a power output of approximately 400 horsepower. At almost the same time, a 435 horsepower nine cylinder radial, built by the Wright Aeronautical Corporation and known as the model "P–2" (Figure 8) successfully completed an endurance run of fifty hours. It would, therefore, appear that there are a number of successful models available, although the Wright model "J" still remains the only American air-cooled aircraft engine, which has been produced in quantities and thoroughly service tested in the air.

Besides Mr. Lawrance, the engineers mostly responsible for the development of the "Wright" air-cooled engines have been Mr. George Mead, for several years Chief Engineer of the Wright Company, and Mr. C. Fayette Taylor, for some time Assistant Engineer and also Chief Engineer of the Wright Company. The "J–5" engine has been developed under the supervision of Mr. E. T. Jones, at present Chief Engineer of the Wright Company.

CONCLUDING REMARKS.

I have in the above recital presented the various stages of a successful development in aircraft engines which is the product of co-operation between Naval Aviation and an efficient and effective private industrial organization. This development has not been accomplished without trouble and the exercise of patience and much physical and mental labor on the part of both the Naval and the Wright Company's representatives. Much of the personnel connected with this development has changed during its progress;

but it seems that a spirit aiming at true accomplishment and a proper evaluation of a government department's relation to private industry in a mutual development has been present. This I believe is in no small measure responsible for the very successful results that have been secured. This development has placed the United States in the forefront as far as the practical production of air-cooled aviation engines is concerned.

This development, initiated mainly for naval purposes, has resulted in the producing of types of engines widely and successfully used in commercial aviation; thus not only the military services but also the aircraft industry has benefited.

In preparing this article I am indebted to the officials of the Wright Company for material and data and in particular to Mr. E. T. Jones, Chief Engineer, Wright Company, and Mr. H. L. Stieber, Resident Navy Inspector at the Wright plant, who have very materially assisted me in the preparation of these notes.

TESTS IN CONNECTION WITH GAS AND METAL ARC WELDING AS APPLIED TO AIRCRAFT CONSTRUCTION.

By Commander H. B. Hird, U. S. Navy, Member.

At the request of the Bureau of Aeronautics of the Navy Department, the Norfolk Navy Yard conducted a series of tests in order to determine certain facts in connection with the use of Gas and Metal Arc Welding as applied to Aircraft Construction. It is believed that these tests will be of considerable interest and an attempt is made herein to cover in sufficient detail the Navy Yard's report to indicate in general the results obtained.

The object of these tests may be itemized as follows:—

(*a*) To determine the thermal effect of gas and metal arc welding as indicated by tensile and bending tests on material.

(*b*) To determine the thermal effect on various gauges of material.

(*c*) To determine the distribution of tension stress in material.

(*d*) To determine the effect of symmetrical deposits of weld metal on tubing.

(*e*) To determine the tensile strength of a tube to which branches are welded.

(*f*) To determine the tensile strength of a tee weld in tubing when stressed at right angles to the tube to which welded.

(*g*) To determine the effect of annealing on gas and metal arc welded tubing.

(*h*) To determine the lightest gauge metal which can be satisfactorily metal arc welded.

(*i*) To determine the size of electrodes, current values, and other elements of technic for the metal arc welding of sheet and tubing.

(*j*) To obtain information for the development of metal arc welding in the Naval Service.

The materials tested were:—

(*a*) Mild carbon sheet steel of various gauges. S. A. E. 1025.
Carbon 0.20—0.30 per cent.
Manganese 0.50—0.80 per cent.
Phosphorus 0.045 maximum per cent.
Sulphur 0.05 maximum per cent.

(*b*) Chrome vanadium sheet steel of various gauges. S. A. E. 6130.
Carbon 0.25—0.35 per cent.
Manganese 0.50—0.80 per cent.
Phosphorus 0.045 maximum per cent.
Sulphur 0.05 maximum per cent.
Chromium 0.80—1.10 per cent.
Vanadium 0.15 minimum per cent.

(*c*) Nickel steel tubing of various diameters and gauges, S. A. E. 2330.
Carbon, 0.25—0.35 per cent.
Manganese, 0.50—0.80 per cent.
Phosphorus, 0.04 maximum per cent.
Sulphur, 0.045 maximum per cent.
Nickel, 3.25—3.75 per cent.

(*d*) Chrome molybdenum tubing of various diameters and gauges. Army Air Service Specification No. 10,231-A.
Carbon, 0.25—0.35 per cent.
Manganese, 0.40—0.60 per cent.
Phosphorus, 0.04 maximum per cent.
Sulphur, 0.045 maximum per cent.
Chromium, 0.80—1.10 per cent.
Molybdenum, 0.15—0.25 minimum per cent.

To obtain the information desired, six tests were conducted as follows:—

No. I.—Tension Tests of Metal Arc Welded Mild, Nickel and Chrome Molybdenum Steel Tubing—Specimens, Sketches A, B, C, D and E, Exhibit (F).

No. II.—Tension Tests of Metal Arc Welded Mild, Nickel and Chrome Molybdenum Steel Tubing—Specimens, Sketches F and G, Exhibit (F).

No. III.—Tension Tests of Metal Arc Welded Mild and Chrome Vanadium Sheet Steel—Specimens H, I, J, K, and L, Exhibit (F).

No. IV.—Bending Tests of Metal Arc Welded Mild and Chrome Vanadium Sheet Steel—Specimens M, N, O, P, Q and R, Exhibit (F).

No. V.—Tension Tests of Gas and Metal Arc Welded Mild, Nickel and Chrome Molybdenum Steel Tubing—Specimens. Sketches A, B and C, Exhibit (F).

No. VI.—Tension Tests of Specimens (Test No. V)—Annealed as per Exhibit (D).

EXHIBIT (D)

TABLE OF ANNEALING TEMPERATURES FOR AEROPLANE TUBING.

(From enclosure (A) of Bureau Aeronautics letter Aer-M-15-FAM 13-17 JJ-44T1 160-39 of no date (August 1925)).

STEEL NO.	#1025	#6150	#2330	CHROME MOLYBDENUM
Place in furnace at temperature not over	1100°F.	1200°F.	1000°F.	1100°F.
Minimum time to heat to annealing temp.	1-1/2 hrs.	2 hrs.	1-1/2 hrs.	2 hrs.
Annealing temperature.	1625°F.	1600°F. to 1625°F.	1450°F. to 1480°F.	1600°F. to 1650°F.
Time to hold tubing at annealing temperature	20 min.	20 min.	10 min.	10 min.
Minimum time to cool 1100°F. in furnace		3 hrs.	2 hrs.	3 hrs.

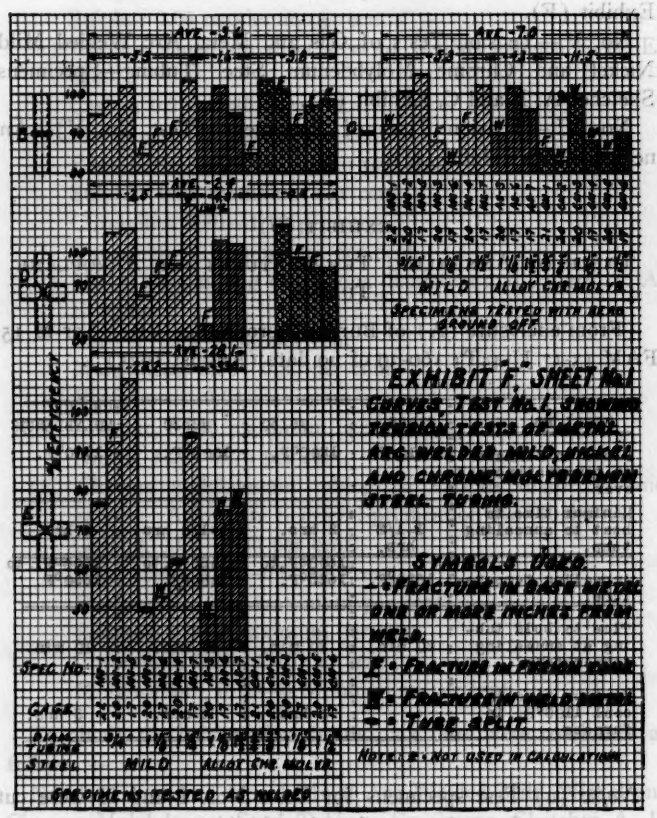
Note: (*a*) Test specimens for Tests Nos. I and II were cut from the same lengths of tubing.

(*b*) Test specimens for Tests Nos. III and IV were cut from the same sheets.

(*c*) Test specimens for Tests Nos. V and VI were cut from the same lengths of tubing.

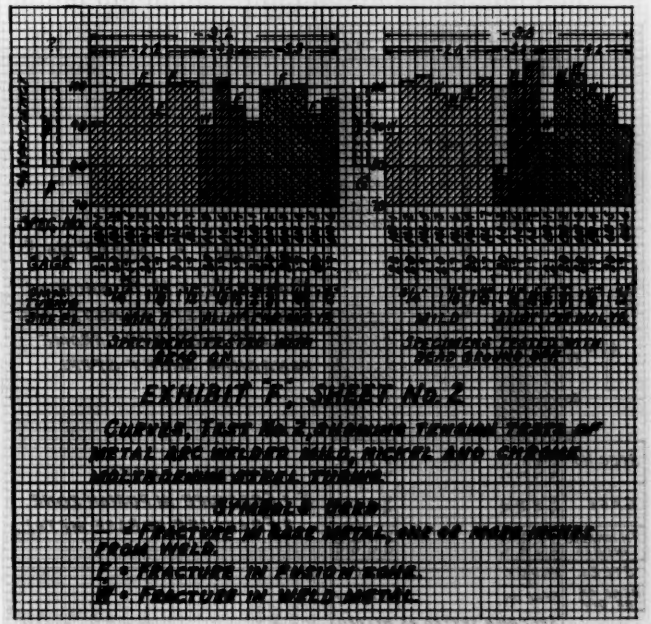
Sketches of the specimens tested, the character of the metal and the diameter and thickness of the tubes are shown in Exhibit (F).

For the metal arc welding a 1000 ampère 60-volt, open circuit welding generator was used, driven by a 220-volt, three-phase



motor. Two standard control panels used for the welding of heavy materials were connected in series to obtain current values, in approximately two ampère steps, ranging from 20 to 50 ampères. The electrode holder was of very light construction, weighing

only one-half a pound and was specially made for these tests. It was found that for work of this class, too dark a lens could not be used, a standard welding number 12 Noviweld lens proved to be too dark and a number 8 lens was found to be quite satisfactory. The electrodes conformed to the Navy Department's



Specifications, Grade A (C–0.06 maximum, Mn–0.15 maximum, P–0.04 maximum, S–0.04 maximum, and Si–0.08). Electrodes of 1/16 inch diameter were used for sheet and tubing of 20, 21 and 22 gauge, and 3/32 inch diameter for sheet and tubing of 17 and 18 gauge.

In connection with the electrodes used, the report contains the following note: 1/16 inch diameter electrodes of two different manufacturers were tried, one bare, the other coated. The coated

wire worked best, especially when the coating was very thin. No coated electrodes of 3/32 inch diameter were available; however, the bare ones used worked satisfactorily. Manufacturers of coated electrodes claim that a softer arc can be obtained with a coated wire and that it is more suitable than bare wire for welding very

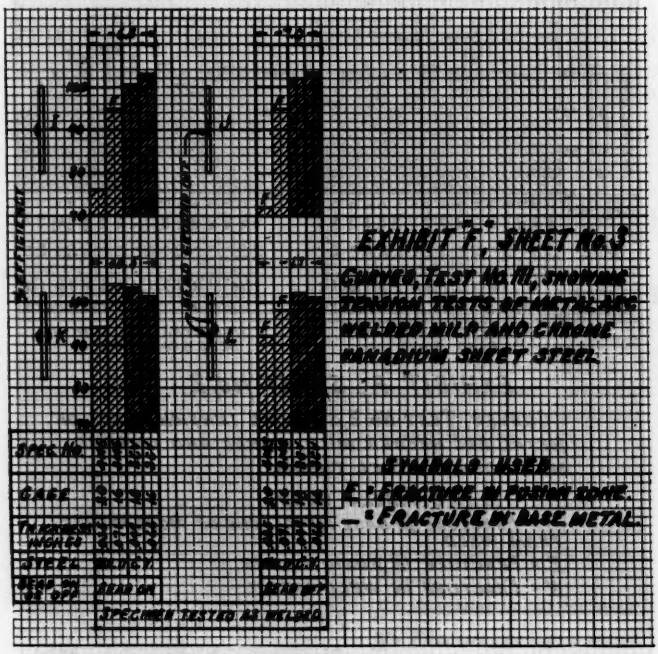


EXHIBIT F, SHEET No. 3
CURVES, TEST No. III, SHOWING TENSION TESTS OF METAL ARC WELDED MILD AND CHROME MOLYBDENUM SHEET STEEL.
EXPLANATION
F = FRACTURE IN FUSION ZONE.
* FRACTURE IN BASE METAL.
SPECIMEN TESTED AS WELDED.

light material. Too much reliance should not be placed upon the claims of manufacturers for their particular coated electrodes, and claims should always be verified by tests. For heavy work the Norfolk Yard uses bare electrodes exclusively.

For the gas welding, a Midget Torch, with a No. 2 tip was used. The welding rod was 1/16 inch diameter, bare, low carbon

steel, Grade E, (Chemical composition the same as for Grade A quoted above for metal arc welding.)

Only one base metal specimen of each size and gauge of tubing, for each class of material welded, was tested. For sheet metal,

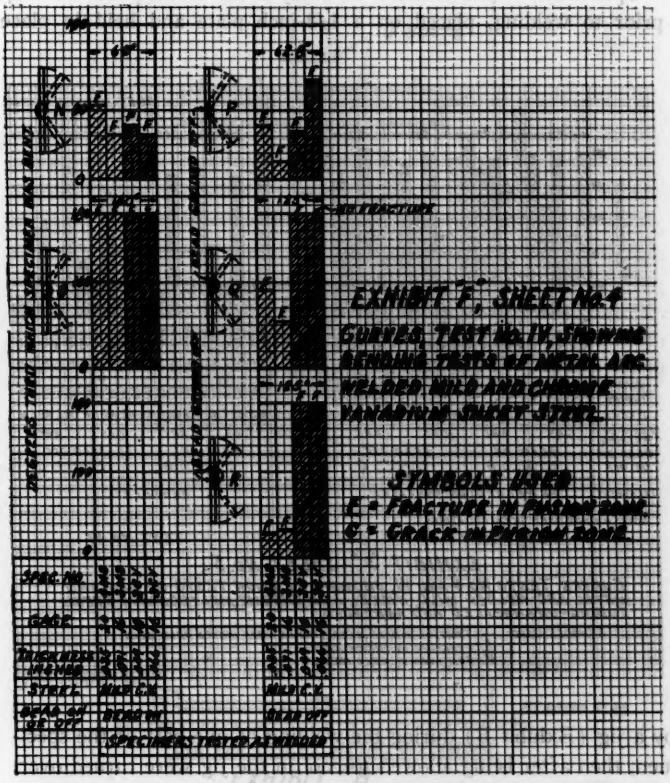


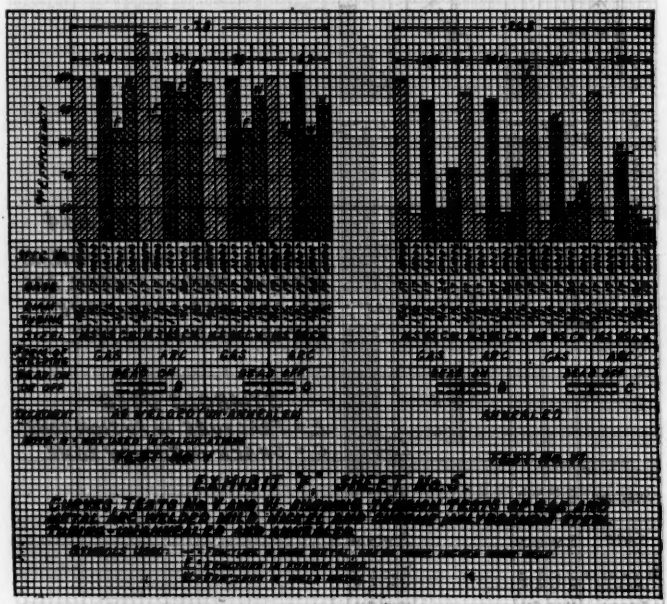
EXHIBIT "F," SHEET No. 4
CURVES, TEST No. IV, SHOWING BENDING TESTS OF METAL ARC WELDED MILD AND CHROME VANADIUM ALLOY STEEL.
SYMBOLS USED
F = FRACTURE IN FUSION ZONE.
C = CRACK IN FUSION ZONE.

only one base metal specimen was tested for each gauge and class of material welded. While it would have been preferable to test three specimens of each, one was considered satisfactory for this test and greatly reduced the number of specimens. Comparisons

for efficiency as shown in curves, Exhibit (F) were related directly to test results of these specimens and not on a basis of pounds per square inch cross section.

Specimens B and C were placed on the bending slab and were gradually turned as welding proceeded.

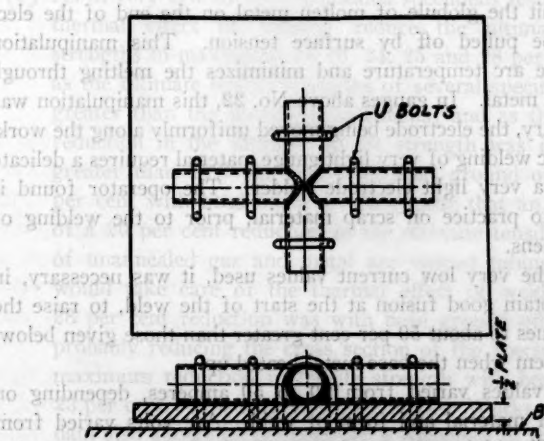
Specimens D and E were carefully fitted in the machine shop prior to welding, placed on a 1/2-inch plate and held in place



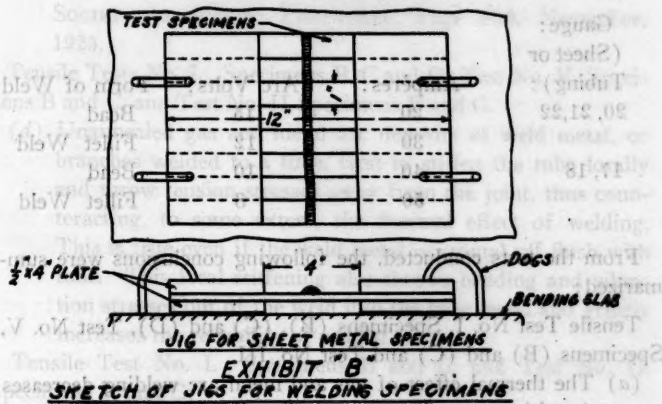
by a jig (Exhibit B) during the welding of one-half of the specimen, then after being allowed to cool in the jig were turned over, reclamped in the jig with liners and the weld finished.

Specimens F and G were clamped to a bending slab during the welding.

Specimens I to L and N to R were cut from the sheet after the deposition of the bead or beads. The sheet was held in a jig (Exhibit B) during welding.



JIG FOR TUBULAR SPECIMENS



JIG FOR SHEET METAL SPECIMENS

EXHIBIT "B"

SKETCH OF JIGS FOR WELDING SPECIMENS

In the deposition of metal with the metal arc on material up to No. 22 gauge, it was found necessary to repeatedly shorten the length of the arc, thus welding with an in and out motion, so as to permit the globule of molten metal on the end of the electrode to be pulled off by surface tension. This manipulation reduces the arc temperature and minimizes the melting through of the base metal. In gauges above No. 22, this manipulation was not necessary, the electrode being moved uniformly along the work.

Metal arc welding of very light gauge material requires a delicate touch and a very light electrode holder. The operator found it necessary to practice on scrap material, prior to the welding of test specimens.

Due to the very low current values used, it was necessary, in order to obtain good fusion at the start of the weld, to raise the current values to about 50 per cent greater than those given below, reducing them when the base metal heated up.

Current values varied from 20 to 50 ampères, depending on thickness of material and form of weld. Arc volts varied from 8 to 15. The higher the ampères used, the lower the arc volts.

Gauge: (Sheet or Tubing):	Ampères:	Arc Volts:	Form of Weld
20, 21,22	20	15	Bead
	30	12	Fillet Weld
17, 18	40	10	Bead
	50	8	Fillet Weld

From the tests conducted, the following conclusions were summarized:

Tensile Test No. I, Specimens (B), (C) and (D), Test No. V, Specimens (B) and (C) and Test No. III.

(*a*) The thermal effect of gas and metal arc welding decreases the ultimate tensile strength of unannealed mild, nickel and chrome molybdenum steel tubing, the per cent reduction in tensile strength being approximately the same for both gas and metal arc welding and for each class of material.

(*b*) In general, the thermal effect decreases as the gauge of the metal increases.

(*c*) As in very few of the unannealed specimens tested did the thermal effect of welding reduce the ultimate tensile strengths to maximums of 20, 24, 25 and 28 per cent; and as the ultimate tensile strengths of several specimens were greater than the unwelded specimens; and as the average reduction in the ultimate tensile strength was in no case greater than 11.5 per cent with bead ground off and 9.0 per cent with bead on, it would appear that an allowance of a 20 per cent reduction in the ultimate tensile strength of unannealed gas and metal arc welded tubing or sheet would take care of the thermal effect of welding. The 28 per cent reduction was with bead ground off, grinding probably reducing the cross section of the specimen. The maximum reduction in tensile strength with bead on was 25 per cent. It is noted that the Fokker Works at Amsterdam also allows 20 per cent for a reduction in the strength of a welded mild steel tube. (JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Page 830, November, 1925.)

Tensile Tests No. I. Specimens B, C and D; Test No. V, Specimens B and C, and Test No. II Specimens F and G.

(*d*) Unannealed gas and metal arc deposits of weld metal, or branches welded to a tube, tend to stiffen the tube locally and throw tension stresses away from the joint, thus counteracting, to some extent, the thermal effect of welding. This is true even if the weld metal is ground off flush with tube. This local stiffening also throws bending and vibration stresses out of the weld into the base metal and greatly increases the reliability of welded joints.

Tensile Test No. I. Specimens B and C, and Test No. II. Specimens F and G.

(*e*) The thermal effect of an unsymmetrical deposit of weld metal on a tube does not affect the ultimate tensile strength of the tube to a greater extent than that of a symmetrical deposit. It is believed that this conclusion will also be true when only one branch is welded to a tube.

Tensile Test No. I. Specimens B and D.

(*f*) The thermal effect of welding two branches on opposite sides of a tube does not affect the ultimate tensile strength of the tube to a greater extent than a circumferential deposit of weld metal. It is believed that this will also be true for more than two tubes.

Tensile Test No. I. Specimen E.

(*g*) The metal arc welding of tubing directly to another tube of the same size and gauge is not considered satisfactory when *branches* are subject to a very high tensile stress, for the tube to which the branches are welded is liable to split at one-half its ultimate tensile strength. This method of attaching a branch would, however, be satisfactory provided the ultimate tensile strength of the branch tubes is one-half or less than that of the main tube.

(*h*) To obtain a tee joint having a high tensile strength at right angles to the tube to which the branch or branches are welded, it will be necessary to reinforce the tube. The best method of doing this is to insert a gusset plate of sheet metal through the tube and extending into the branch or branches, as shown in Exhibit C.

Tensile Test No. VI.

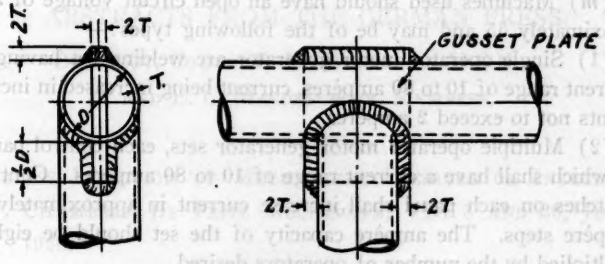
(*i*) The annealing of gas and metal arc welded, mild, alloy and chrome molybdenum steel tubing, decreases its ultimate tensile strength approximately three times that of unannealed tubing similarly welded, the reduction in tensile strength being as much as 46 per cent. Welded tubing should, therefore, not be annealed.

(*j*) Mild, nickel and chrome molybdenum steel tubing as light as No. 20 gauge can be satisfactorily gas or metal arc welded.

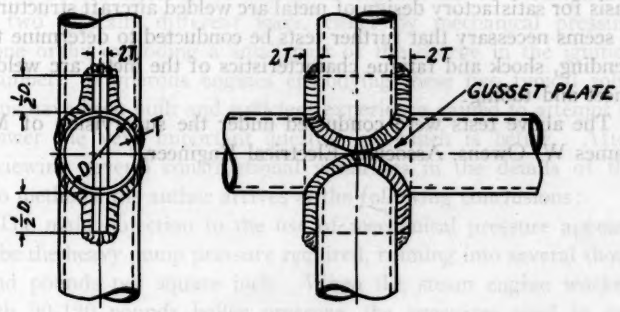
Bending Test No. IV. Specimens N and M.

(*k*) Sheet metal having a weld on one side, if bent so as to put the weld under compression, can be bent through 360 degrees before crack occurs, whereas when bent so that weld is on tension side, a crack might occur at about a 50-degree bend. It is not believed that in light gauge material the composition of the material will materially affect the angle of bend.

(*l*) Metal arc welding is equally as satisfactory as gas welding for aeroplane construction and has the decided advantage that



TEE CONNECTION



CROSS CONNECTION

NOTE:

NO TESTS HAVE BEEN MADE TO DETERMINE THICKNESS AND SIZE OF GUSSET PLATE, HOWEVER IT IS BELIEVED THAT PROPORTIONS GIVEN WILL BE SATISFACTORY

EXHIBIT "C"

TYPICAL SKETCHES OF RECOMMENDED CONNECTIONS FOR TUBING.

warping is much less than that produced by gas welding, and that joints can be satisfactorily welded in a jig.

(*m*) Machines used should have an open circuit voltage of approximately 55 and may be of the following types:—

(1) Single operator motor generator arc welding set having a current range of 10 to 80 ampères, current being increased in increments not to exceed 2 ampères.

(2) Multiple operator motor generator sets, each control panel of which shall have a current range of 10 to 80 ampères. Control switches on each panel shall increase current in approximately 2 ampère steps. The ampère capacity of the set should be eighty multiplied by the number of operators desired.

In making recommendations based upon the above conclusions the report states that while the results of these tests will form a basis for satisfactory design of metal arc welded aircraft structures, it seems necessary that further tests be conducted to determine the bending, shock and fatigue characteristics of the metal arc welded sheet and tubing.

The above tests were conducted under the supervision of Mr. James W. Owens, Associate Electrical Engineer.

ABSTRACTS FROM THE GERMAN PRESS.

By Lieut. Comdr. E. W. Burrough, U.S.N.
and E. C. Magdeburger, Members.

Airless Injection by Mechanical Pressure or Injection Chambers, By Fritz Modersohn, VDI 5 and 26, June, 1926.

The airless injection of fuel in oil engines can be accomplished in two typically different ways, either by mechanical pressure alone or by exploding a small part of the charge in the ignition chamber. Numerous engines embodying these two typical solutions have been built and sufficient experience gained to attempt to answer the very important question of which is better. After reviewing several constructional variations in the details of the two methods, the author arrives at the following conclusions:

The main objection to the use of mechanical pressure appears to be the heavy pump pressure required, running into several thousand pounds per square inch. When the steam engine worked with 90-120 pounds boiler pressure, the pressures used in gas engines (300-375 pounds) appeared to render questionable the reliability of the crank mechanism. The high compression Diesel engine carried these pressures still higher, up to 500-550 pounds, which also caused anxiety at the start. Today steam engines working with steam pressures of 600-900 pounds per square inch are in the experimental stage, and it really makes but little difference whether the pump pressure is 1000 pounds or 5000 pounds per square inch; the question being solely one of practical difficulties. Experience to date indicates that there are none. It is only natural, of course, that such high pressures must be met by a proper design of pump and piping, but it is no more difficult to build a pump for 5000 pounds pressure than it is for 1000 pounds,

provided that cams, rollers and pins are liberally dimensioned and the pump drive connected to forced feed lubrication to keep down the wear.

Furthermore, it is said that mechanical injection usually requires very small fuel valve passages, which are quite beyond any existing machine shop practice and which may easily clog up. It is

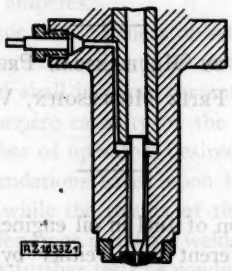


FIG. 1.—FUEL VALVE OF THE CLOSED TYPE.

quite true that, so far, there is a limit as to the size of hole which may be drilled without resorting to watch makers' tools (about .012 inch). However, .008 inch holes which have an area of less than one-half of .012 inch holes, have already been used. When the fuel is carefully filtered, which is easily done with existing means, no occasion to fear clogging arises even with fuels of high ash content.

As an objection to the ignition chamber, it is often said that the chemical processes within the chamber are not sufficiently clear, as compared with those resulting from mechanical injection. Although further study of the processes involved is highly desirable from a theoretical point of view, the practical success of the injection chamber type of oil engines has long disposed of any such apprehensions.

It is said that the ducts connecting the ignition chamber with the interior of the working cylinder burn out and require the replacement of the whole chamber. This is merely incidental to certain not fully developed designs, using thin walled chambers with a large number of small holes comparatively close together, with the bridges between the holes not well enough cooled, and cannot by any means be attributed to the type as a whole.

These are the objections most often raised, but they are well disproved by the reliability of both methods of operation.

The fuel consumption is usually slightly better for the engines using mechanical injection, although the injection chamber type gives equally favorable results in rather large cylinders. So far only one large engine has been built using injection chamber, namely, a 4 cylinder 1000 B.H.P. Körting engine, with injection chamber, as shown on Figure 2, whereas no mechanical injection has been as yet applied to cylinders of such large capacity,* which may therefore show greater improvement.

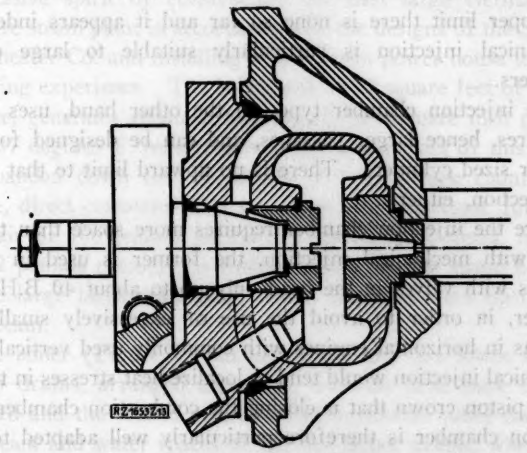


FIG. 2.—IGNITION CHAMBER OF KORTING ENGINE.

On the other hand, the injection chamber is superior to mechanical injection when using heavy, slow igniting and slow burning fuels. Even in small injection chamber engines, tar oils may be used without light ignition oil and without any change from the ordinary equipment even on light loads and no load. Oils with high ash content and high viscosity may also be used after these latter are warmed up. With mechanical injection as well as in the old air injection type of engine, the light injection oil to precede the main charge is so far unavoidable.

* The author evidently refers to German engines only, since both Vickers and Doxford have built much larger cylinders; the latter with fine fuel economy.

Superior speed regulation as compared with air injection type of engines is characteristic of both types of airless injection, while particularly low speeds (30 R.P.M. and below) are obtainable in multi-cylinder engines with mechanical injection.

Hence the two methods must be conceded to be equivalent technically, and their use is governed essentially by their adaptability to the size and type of engine.

Since mechanical injection is limited by the size of hole through the nozzle plate (about .012 inch), its lower limit appears to be a cylinder of 30-40 B.H.P. at the usual R.P.M., although smaller units have been built, such as the M.A.N. truck motor. As to the upper limit there is none so far and it appears indeed that mechanical injection is particularly suitable to large capacity cylinders.

The injection chamber type, on the other hand, uses smaller pressures, hence larger passages, and can be designed for much smaller sized cylinders. There is no upward limit to that method of injection, either.

Since the injection chamber requires more space than the fuel valve with mechanical injection, the former is used in vertical engines with valves in the head only up to about 40 B.H.P. per cylinder, in order to avoid the use of excessively small holes, whereas in horizontal engines with commonly used vertical valves mechanical injection would tend to localize heat stresses in the part of the piston crown that is closing the combustion chamber. The injection chamber is therefore particularly well adapted to horizontal engines up to the largest sizes.

Two cycle engines using crank case compression are universally fitted with injection chambers, whereas on larger sizes where separate scavenging pump is provided either method may be used.

E. C. M.

HIGH PRESSURE STEAM INSTALLATIONS IN EUROPE. V.D.I., 22 MAY, 1926.

The competition of the oil engine with its high thermal efficiency is directly responsible for the development of means of improving the efficiency of steam installations, and the high com-

pression pressures employed pointed the way toward greater efficiency with steam. Today high pressure steam is not merely discussed theoretically, but various engineering organizations have actually constructed and are operating such plants both here and in Europe. Furthermore, high pressure steam will be given a trial shortly on a merchant ship in England, hence the two continental installations described in the recent issue of V.D.I. devoted to "steam" are deemed of interest to naval engineers.

(*a*) *Tests of the A. Borsig Co.'s 850-pound steam installation, by Prof. E. Josse.* This well known engineering firm proved its progressive spirit by constructing the first large German high pressure steam plant in accordance with the designs of the Schmidt Superheater Co. and installing it in its own power house to obtain operating experience. The boiler with 3012 square feet of heating surface, generates steam at 853 pounds per square inch pressure and 797 degrees F. temperature. The larger part of this steam* is expanded down to 142 pounds in a two-stage counter flow engine, direct connected to a two-stage air compressor furnishing shop air of 100 pounds pressure. The exhaust steam, normally of 142 pounds pressure, is further utilized in the steam hammers of the large blacksmith shop, which formerly were operated by live steam.

The boiler (Figure 1) has two upper and two lower seamless drawn drums, arranged crosswise, with the first bank of tubes inclined and the second vertical, with horizontal tubes connecting the steam and water spaces of the two upper drums, whereas the lower drums are not connected, so that the tubes may expand independently of each other. It will be noted that the points of entry of the tubes into the drums are protected from the influence of the flue gases, and water circulation between upper and lower drums is maintained through special tubes at the ends of the drums that are also protected from the excessive heat. A superheater is installed between the upper ends of the high pressure banks of tubes. Flue gases leaving the second bank of tubes are further cooled by a 28.4 pound tubular boiler installed immediately behind the high pressure boiler, which serves as a preheater only, to bring the water to about 212 degrees F.

* The boiler being designed for an ultimate turbine installation of larger size.

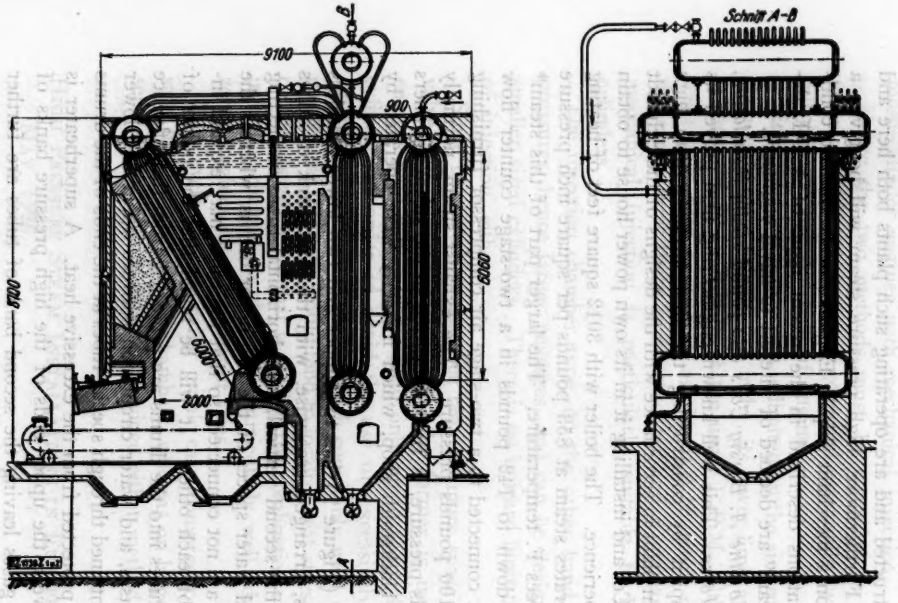


FIG. 1.—BORSIG-SCHMIDT BOILER FOR 850 POUNDS PRESSURE AND 800 DEGREES F., SUPERHEAT.

Heating surface of high pressure boiler	3012 square feet.
Heating surface of first bank of tubes	1527 square feet.
Heating surface of second bank of tubes	1485 square feet.
Heating surface of the superheater	1464 square feet.
Heating surface of the 30-pound preheater boiler	2150 square feet.
Grate surface	110 square feet.

The engine consists of two single acting cylinders arranged in tandem, as made by the Schmidt Superheater Co., and it developed on test 844 indicated horsepower. The three indicator cards taken on the steam header immediately before the high pressure valve, on the high pressure cylinder and on the low pressure cylinder, are shown on Figure 2.

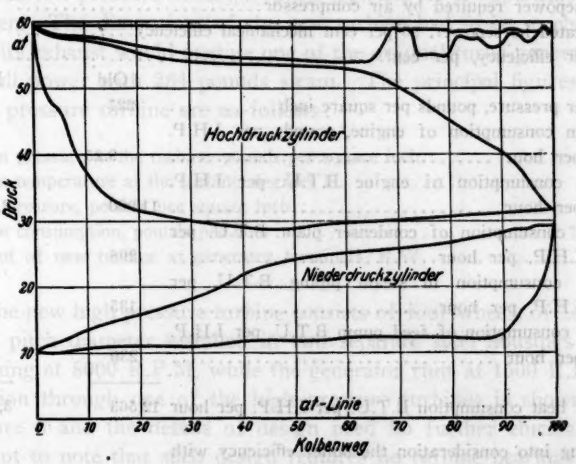


FIG. 2.—AVERAGE INDICATOR DIAGRAMS BROUGHT TO A COMMON SCALE.

Druck—pressure, Hochdruckzylinder—H. P. cylinder.
Niederdruckzylinder—L. P. cylinder.
At—Linie—atmospheric line.
Kolbenweg—per cent of stroke.

Results of tests showed a boiler efficiency of about 83 per cent, the heat absorption being distributed as follows: Preheater, 7.7 per cent; High Pressure Boiler, 62.3 per cent, with 54.7 per cent in the first tube bank and 7.6 per cent in the second, and Superheater 12.9 per cent. Only 7.07 pounds of water per square foot of heating surface per hour was evaporated, although 7.9 was evaporated on a previous official test. A steam consumption of 16.7 pounds per indicated horsepower was recorded, with a thermal efficiency of 91 per cent for the whole engine, with 92.7 per cent for the H.P. and 88.5 per cent for the L.P. cylinder.

The economy of the new installation as compared with the old, using 227.2 pounds steam pressure in a condensing engine for driving the air compressor and throttling the steam for the steam hammers up to 142 pounds, assuming 13,860 pounds of steam per hour for that purpose, is evident from the following table:

Steam used in steam hammers, pounds per hour		13,860
Horsepower required by air compressor		760
Indicated horsepower, 90 per cent mechanical efficiency		844
Boiler efficiency, per cent		80
	Old	New
Boiler pressure, pounds per square inch	227	853
Steam consumption of engine, pounds per I.H.P. per hour	9.25	16.7
Heat consumption of engine B.T.U. per I.H.P. per hour	11,900	2,580
Heat consumption of condenser plant B.T.U. per I.H.P. per hour	298	0
Heat consumption in steam piping B.T.U. per I.H.P. per hour	135	151
Heat consumption of feed pump B.T.U. per I.H.P. per hour	230	548
Total heat consumption B.T.U. per I.H.P. per hour	12,563	3,279
Taking into consideration the boiler efficiency with 12 per cent added for loss in starting and due to coal in ashes, total B.T.U. per I.H.P. per hour.	17,588	4,590

Thus a large saving of 17,588—4,590 = 12,998 B.T.U. per I.H.P. per hour is recorded. The heavier first cost of the high pressure installation at prevailing German prices can be paid in a year and a half out of the savings.

(*b*) *Steam turbines for high pressures, by N. G. Noack.* The author describes new tendencies in turbine design as brought out by the well known Brown, Boveri & Co. The following description of the first high pressure turbine installation in Europe is taken as being of unusual interest.

The installation was made in the power house of Centrales Electriques des Flandres Langerbrugge, Belgium, where the Brown Boveri Co., during the years 1919-1925 had already installed four

turbo generators of 6000 K.W. working with steam pressure of 284 pounds. To improve the efficiency of the installation, the steam pressure was raised to 720 pounds and the pressure difference thus created used in a separate turbo-generator unit. To avoid superheating between stages and yet receive exhaust steam from the high pressure, equivalent to steam generated in the original low pressure boilers, a rather high degree of superheat was chosen. The dimensions of the new turbine had to be so chosen that its exhaust would operate one of the original turbo-generators at full power with 284 pounds steam. The principal figures for high pressure turbine are as follows:

Steam pressure at the turbine, pounds per square inch	720
Steam temperature at the turbine, degrees F	824
Back pressure, pounds per square inch	298
Steam consumption, pounds per hour	79,200
Output of new turbine at generator terminals, K.W	1675

The new high pressure turbine consists of four wheels of 14.65-inch pitch diameter installed in two separate steel housings and running at 8000 R.P.M. while the generator runs at 1500 R.P.M. Section through one of the high pressure turbines is shown on Figure 3 and the details of design need no further elucidation, except to note that such design requires no turbine bearings and no high pressure stuffing boxes and permits free expansion of the housing. For similar reasons the steam mains were made of several small diameter pipes bent U shape. For test purposes the installation was operated with steam of 1058 degrees F. temperature and showed no signs of weakness, although it is not intended to subject it again to such severe stresses.

The nozzles and blading were milled from solid stock and inserted separately. The end cover was so proportioned that the nozzle ring was continuous.

In this installation steam was also largely used for water heating, which is brought up to 385 degrees F. by the exhaust from the high pressure turbine.

The three high pressure boilers for this plant were furnished by the Babcock & Wilcox Co. (London), for which the following data are given:

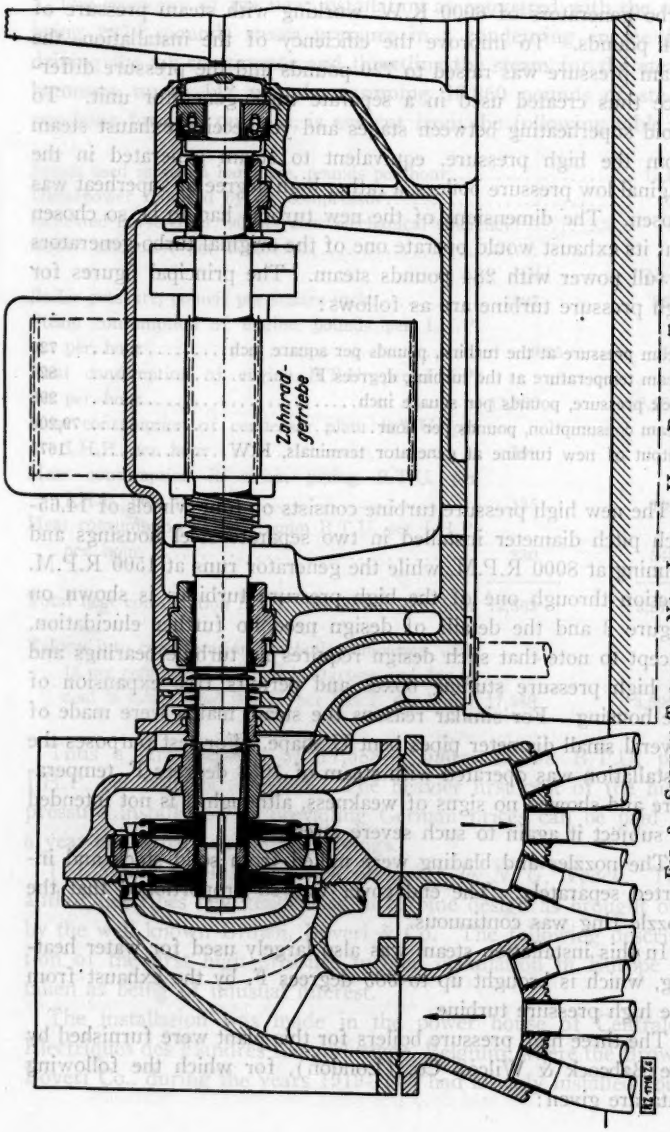


FIG. 3.—SECTION THROUGH NEW INITIAL HIGH PRESSURE TURBINE.
Zahnrad getriebe—Gear drive.

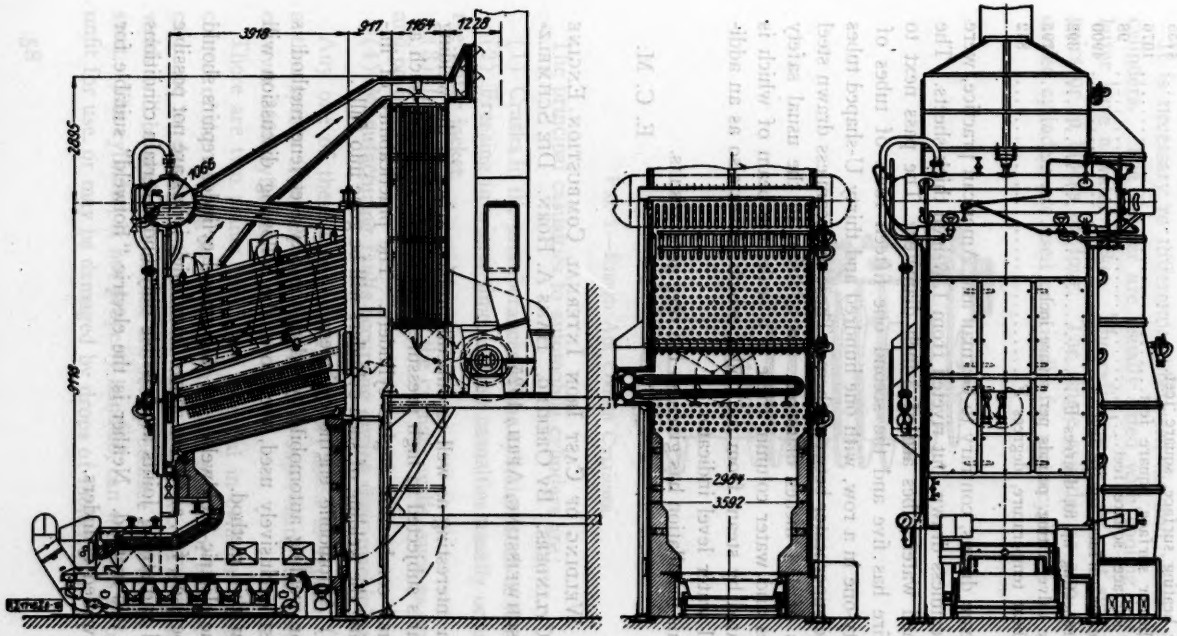


FIG. 4.—B. & W. BOILER OF 4750 SQUARE FEET OF HEATING SURFACE FOR 800 POUNDS PRESSURE AND 840 DEGREES F. SUPERHEAT.

Boiler heating surface, square feet	4755
Superheater surface, square feet	1076
Grate surface, square feet	98
Air heater surface, square feet	4900
Air preheating up to, degrees F	392
Safety valve setting, pounds per square inch	795
Superheat temperature, degrees F	842

Steam drums, contrary to German and American practice, were not seamless drawn, but riveted from 1.57-inch thick sheets. The inclined water tubes are of 3-inch diameter. The pass next to the fire has five and the second one fifteen rows of tubes of twenty-one in a row, with one hundred and thirty U-shaped tubes of the superheater between them. These are seamless drawn steel tubes of 1½ inches diameter. In addition to the usual safety valves and water column, there is a scale, the beam of which is operated by steam from the boiler, which serves also as an additional water level indicator.

This installation has given so far very good results.

E. C. M.

THE WELDING OF CAST IRON INTERNAL COMBUSTION ENGINE CYLINDERS. BY OBERING. HANS A. HORN. DIE SCHMELZSCHWEISSUNG, APRIL, 1925.

An interesting as well as difficult problem arises in the welding of parts subjected to high pressures and severe vibration, such as internal combustion engine cylinders. The practicability of improving the welding of such cylinders will be gone into fully.

(*a*) Automobile engine cylinders.

For welding automobile cylinders the oxy-acetylene method is almost exclusively used, therefore, the following discussion will be on this method.

The electric "coldweld" for thin walled cylinder repairs should not be used, for, as will be shown later, thick joints are not possible at all and strong joints are possible only under certain conditions with this method. Neither is the electric "hotweld" suitable for thin walled cylinders.

It is necessary to differentiate between air and water cooled cylinders. The former are usually fitted with cooling fins, the latter have a water jacket. The cylinders can be cast singly, in pairs or in blocks of four. The single cylinder as also the air cooled cylinders are used principally for motorcycles.

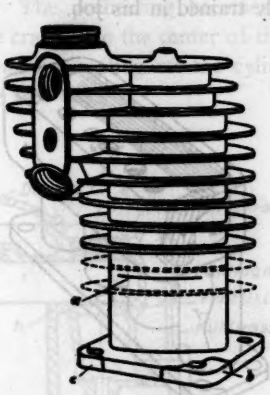


FIG. 1.—RIBBED AIR COOLED CYLINDER.

The principal casualties to automobile cylinders are:

(1) Cracks in the protuberances of the cylinder, such as flanges.

(2) Freezing in winter time. These casualties usually occur in the water jacket.

(3) Injuries to the inside of the cylinder walls, such as cracks and porous spots.

(4) Manufacturing faults occurring during their machining.

Any one of these injuries may be very hard to repair. The author speaks from his past experience in supervising the welding of over 3000 cylinders.

There are two points which are essential in cylinder welding: control of the welding so as not to introduce strains or tension and care not to damage any threads, valve seats, etc. The best of welds will be useless if strains are set up because new cracks will start. The cylinder, if not properly handled, can warp so as to be unfit for use, or may be damaged by drops of molten iron, so that

the old saying, "The operation was a success, but the patient died" is very applicable. The weld must be not only tough and dense, but sufficiently soft to be filed off.

Cementing defects in the weld with iron cement as well as filling up porous spots is unsatisfactory and deceptive. The welder must be fully trained in his job.

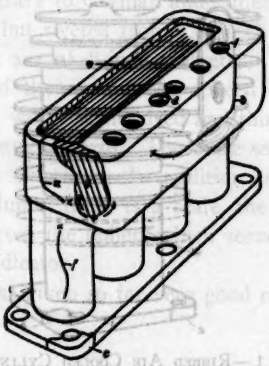


FIG. 2.—CYLINDER BLOCK.

It will usually be found necessary to heat the casting before and after welding to remove strains. This may be done in gas, electric or charcoal muffle furnace. The latter is very good. Preheating with the flame of the welding apparatus is unsatisfactory, unsafe and has the danger of allowing uneven heating and too rapid cooling, which may cause strains resulting in cracks. The castings should be allowed to cool slowly in hot ashes or hot sand.

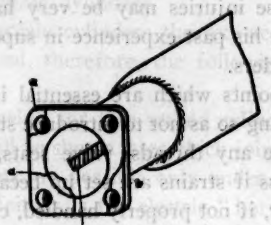


FIG. 3.—SINGLE CYLINDER WITH WATER JACKET.

A good weld will result if the casting is red hot before weld is started. A red hot casting will have no internal strains and as the melting temperature of the welding rod will thus be only a few hundred degrees higher than that of the casting, a much better fusion of metals will result than with a casting that has been insufficiently heated. The preheating becomes of greater importance the nearer the crack is to the center of the casting.

The length of time for cooling off the cylinder depends on its mass and may take two days.

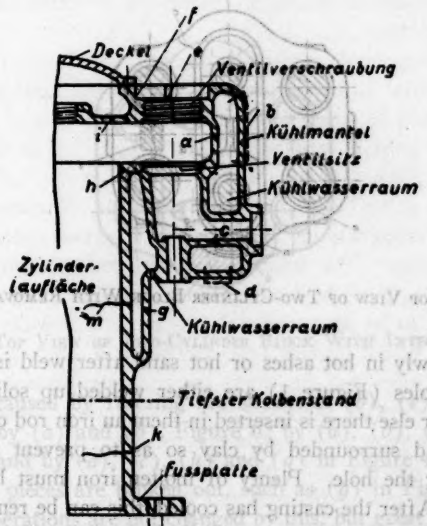


FIG. 4.—CROSS SECTION OF AUTOMOBILE 4-CYCLE CYLINDER.

All parts of the cylinder which are removable should be removed before the work of welding is begun. Parts which cannot be removed, but might be damaged, such as screw threads, valve seats, etc., should be protected with wet asbestos, clay or threaded plugs.

With the help of the sketches herewith, we will now take up the welding of cracks.

In Figure 1—air cooled cylinder with fins—the bottom flange is broken at (*b*) and (*c*) Both breaks are through the bolt holes. Similar cracks in the bottom flanges are shown in Figures 3 and 4.

All of such cracks can be repaired without preheating, but great care must be taken that the main part of the cylinder is not unduly heated during the welding. Hold the welding flame toward the outside, cover the cylinders with wet clay, leaving only the part to be welded free and open. Above all, let the welded portion

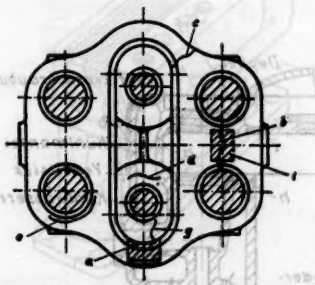


FIG. 5.—TOP VIEW OF TWO-CYLINDER BLOCK WITH REMOVABLE HEAD.

cool off slowly in hot ashes or hot sand after weld is completed. The bolt holes (Figure 1) are either welded up solid and later bored out or else there is inserted in them an iron rod covered with graphite and surrounded by clay so as to prevent molten iron from filling the hole. Plenty of molten iron must be placed at the weld. After the casting has cooled, this can be removed by an electric grinder or file. An old file or scraper can be used to spread out the fluid metal. It is most important that the broken pieces be accurately placed before welding is begun. Tiny pieces are not put on, but new metal melted on instead.

If the cracks are close to the cylinder walls, pre- and after-heating are essential.

The above remarks on cracks in flanges apply also to those in circulating water pipes and for inlet and exhaust cages and lines. In all cases where preheating is not used, the heating incident to welding must be localized.

The work becomes much more difficult when cracks occur in the cylinder or water jacket. Cracks in the water jacket occur almost exclusively from freezing, such as in winter months when water is not drained off while engine is idle. For several reasons cracks from freezing, especially in the jackets, are easier to weld than those from excessive heat. Regardless of location, form or length of cracks, the whole cylinder should be preheated before welding. No exceptions to this rule should be allowed, no matter how simple the crack may be. Violation of this rule will cause trouble in 95 per cent of the cases.

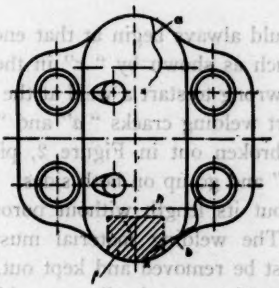


FIG. 6.—TOP VIEW OF TWO-CYLINDER BLOCK WITH INTEGRAL HEAD.

Cracks caused by freezing are shown at (*b*), (*c*) and (*e*) in Figure 5, by (*a*) and (*b*), Figure 6; by (*a*), (*b*), (*e*) and (*d*), Figure 2, and by (*a*), (*c*), (*h*) and (*g*) in Figure 4. Occasionally whole pieces are broken out, such as (*g*) in Figure 2. The welding operations are not changed by this, but great care must be taken to place the broken piece in its correct position. If necessary, take cuts on the broken piece so there will be no parts jutting out.

First of all, examine carefully for all cracks, as it frequently happens that a crack will be welded and then after the job is completed it is found that others were overlooked. The best test for cracks is by hydraulic pressure. This is done by blank flanging or putting wooden plugs in all openings. A pressure of five atmospheres will be sufficient to discover cracks. Just because the

preparations and tests may take more time than the actual welding is no reason for omitting them. Any cracks found by this method, if they are hard to see, are marked.

If several cracks are found in the same cylinder, they can be done with one preheating of the cylinder if they are small and the cylinder does not cool off unduly during the operation. Otherwise the welding must be done in stages and the cylinder put back in the muffle oven when it begins to cool.

The frequency of the interruption in the welding is contingent upon the extent of the work. Even a single crack, if very large, may require several preheats. Some cracks extend all the way around a cylinder.

The welding should always begin at that end of a crack which lies in the metal such as shown by "*x*" in the figures. For example, it would be wrong to start a weld at the hole of crack "*b*," Figure 2, or to start welding cracks "*a*" and "*e*" at the top. In welding the piece broken out in Figure 2, piece "*g*," the weld should begin at "*x*" and go up on both sides. The crack must be well filled throughout its length without porous spots; to spare metal is wrong. The welding material must be fluid. Rust, scale, dirt, etc., must be removed and kept out. Great care must be taken not to let molten metal adhere to thin walls in cooling space, which would prevent free circulation of cooling water.

It is not always that the cracks are so accessible and frequently considerable work in preparation is necessary before the welding is begun. To show an example of this, see cracks "*a*," "*i*," "*c*" *and* "*h*" in Figure 4, and "*a*" in Figure 5. To weld "*a*," Figure 4, the opposite piece "*b*" must first be removed; to weld "*c*" the piece "*d*" must be removed, etc. To weld "*g*," Figure 5, piece "*a*" must first be removed.

After the location and extent of the cracks have been determined and made plainly visible with paint or chalk, the necessary piece of cylinder which must be removed to do the welding is taken out. This should be done in the easiest way—by boring, cutting or sawing, but *not* by burning. The whole cylinder is then uniformly heated. The welding of the cracks must then be done rapidly and if possible the piece removed to facilitate the work should be welded back in the same heat. Otherwise the

cylinder must be reheated before piece is welded back. The cylinder should then be carefully and slowly cooled. If, while finished job is tested under pressure, any porosity appears, then it is most essential that the broken piece be removed and the whole job done over again. Good welding is most essential.

The pressure test is always made after smearing up the welded spots, so as to be sure they will all be noted.

E. W. B.

NOTES.

THE HIGH PRESSURE TURBINE STEAMER *KING GEORGE V.*

The *King George V*, recently built by William Denny & Brothers, Ltd., Dumbarton, Scotland, which has now completed very satisfactory trials on the Clyde, has aroused a considerable amount of interest in the shipbuilding and engineering world. The most noteworthy departure in the vessel from former practice consists of the installation by The Parsons Marine Steam Turbine Co., Ltd., of high pressure geared turbines in place of the former lower pressure direct driven ones.

The working pressure is 550 pounds per square inch. Steam is supplied by two high pressure watertube boilers of the Yarrow type.

A complete description of the machinery follows, but attention may be drawn here to a few outstanding features in the design of the vessel herself. She is mid-way in size between the *King Edward* and the *Queen Alexandra*, being 260 feet long by 32 feet beam by 11 feet depth. Externally, she is readily distinguishable from the other vessels of Messrs. Turbine Steamers' fleet, by the closed sides between the promenade and shade decks, the whole of this space having been converted into a fully covered-in promenade. A large number of the windows at the sides are of the opening type and it is considered that this closing in will probably be one of the most appreciated innovations in the present vessel. A somewhat similar scheme has already been carried out by the same builders on a few cross-channel steamers and has been found of the greatest benefit to the passengers in almost all kinds of weather.

NEW FEATURES IN THE ACCOMMODATIONS.

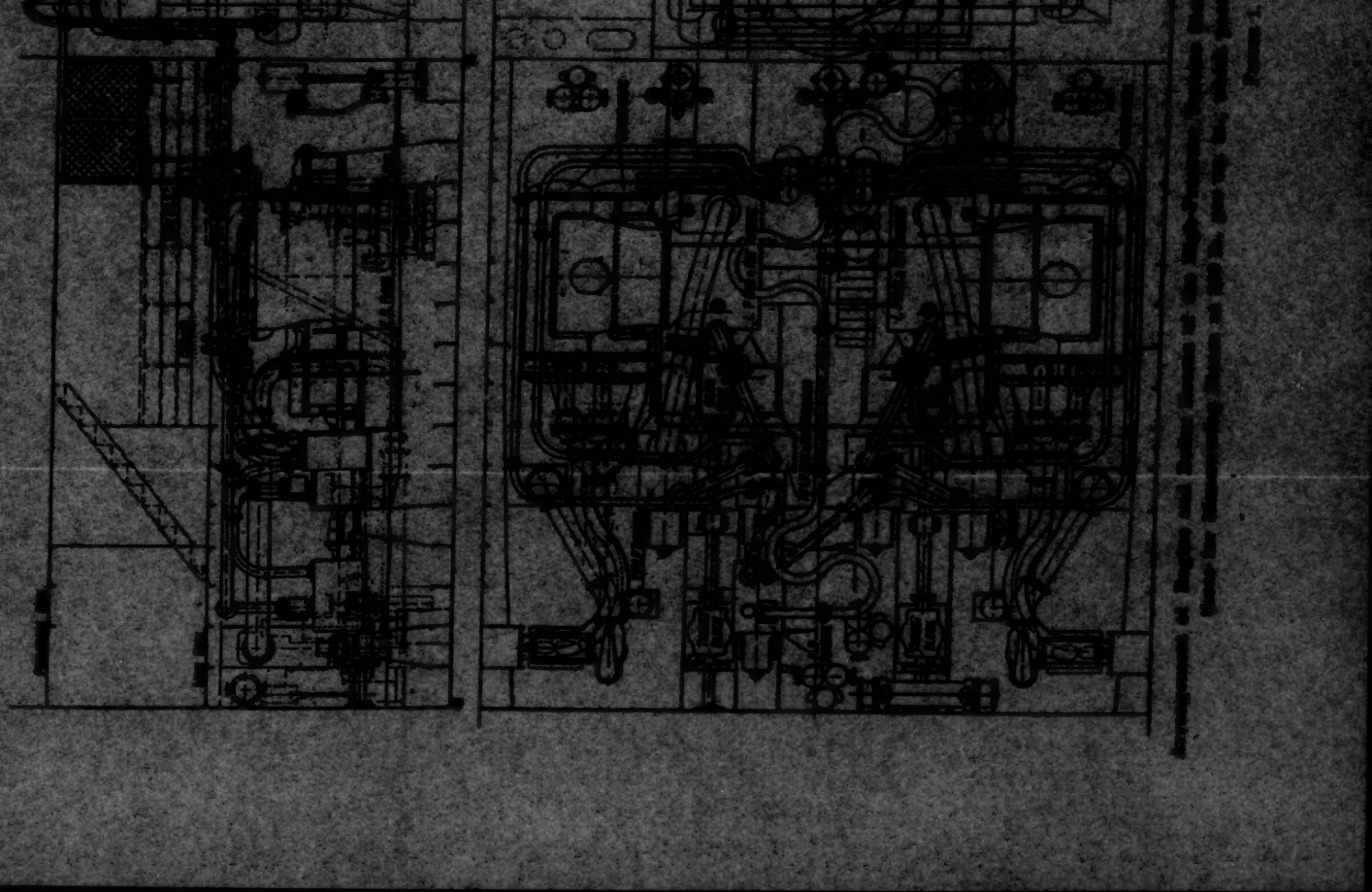
The other radical departure from the earlier vessels is found in the altered position of the dining saloon which has now been arranged on the main deck aft instead of on the lower deck. This saloon has a large seating capacity and is very attractive in appearance, being paneled in mahogany, and fitted with very large windows at the sides, permitting an uninterrupted view of the scenery to the occupants.

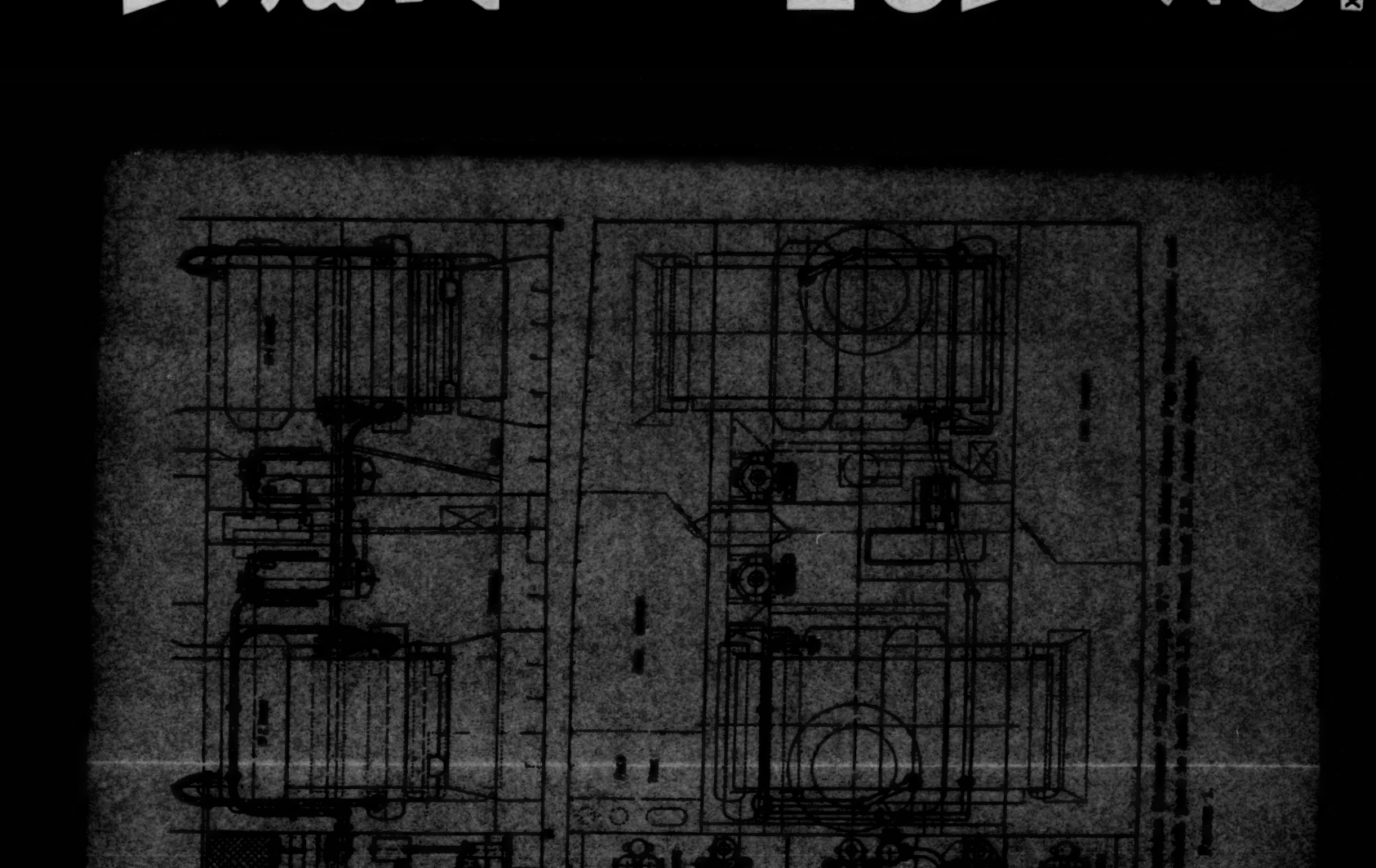
The space on the lower deck formerly occupied by the dining saloon has now been utilized as a tea room and lounge, while a smoking room, tastefully finished in oak, has also been fitted up.

Throughout the whole of the first-class saloons and lavatory accommodations a special system of fan ventilation has been arranged, thereby preventing any suggestion of stuffiness which is sometimes met with below decks.

While the alterations in the steerage accommodations are not so radical as those in the saloon, considerable improvements have also been effected here, a portion of the covered-in- promenade deck has been allocated to their use and improved arrangements can be seen in the large dining saloon and smoking rooms.

A large number of different trials have been carried out in order to thoroughly test Messrs. Parsons' latest development in machinery and on the





Cloch to Cumbrae run recently, the mean speed obtained was well in excess of the contract requirements. It is confidently anticipated that this vessel will prove an even greater attraction than the earlier turbine steamers for the same owners and that she will soon become a great favorite with the traveling public.

DESCRIPTION OF THE MACHINERY.

The distinguishing feature of the machinery in this vessel, as previously stated, is that it is designed for high steam pressure, the boiler pressure being 550 pounds per square inch.

The propelling machinery consists of a set of three turbines geared by single reduction to the propeller shafts, such as has been frequently adopted in previous geared turbine ships. In this installation, however, there is an additional high pressure turbine designed to utilize the expansion of the steam from 500 pounds to about 200 pounds. This high pressure turbine is connected in tandem with the first turbine on the port shaft, and in order to give equal power on the shafts, the main turbines are so designed that a larger proportion of the steam from the high pressure turbine will pass to the starboard shaft. The disposition of the turbines can be clearly seen from the illustration of the main machinery arrangement, Figure 1.

Astern turbines are provided, a high pressure astern turbine in each of the first intermediate ahead turbines and a low pressure astern turbine in the casing of the low pressure ahead turbine in the usual manner.

It will thus be seen that when steaming ahead, the turbines utilize the full expansion of the steam from 500 pounds to condenser vacuum, while for maneuvering ahead or astern, the two shafts can run independently with a pressure of 200 pounds. Nozzles are fitted in the maneuvering steam pipes to reduce the steam pressure to this value.

The steam is superheated to a total temperature of 750 degrees F.

The operation of the machinery is controlled from the starting platform by three valves, a full power valve admitting steam to the high pressure turbine for continuous ahead running and two maneuvering valves which admit steam either ahead or astern to either side. Automatic quick closing valves are fitted in the exhaust pipes between the high pressure ahead turbine and the first intermediate pressure turbines on either side.

The total shaft horsepower developed on the two shafts is 3500, at a speed of 570 revolutions per minute. The high pressure turbine and the first two turbines of each shaft run at 6000 revolutions per minute and the low pressure turbines at 3000 revolutions per minute. Of this total power the high pressure turbine develops 550 shaft horsepower.

TURBINE BLADING.

The ahead blading is of the reaction type throughout, and in view of the high pressure, the blades are necessarily short. Leakage, however, is minimized by the use of end-tightened blading in the high pressure turbine and in the first two turbines of each side. Axial clearance of the blading is controlled in the longitudinal direction by the thrust block, while large radial clearances are provided over the blade tips. The clearance necessary for running is thus preserved, notwithstanding any possibility of slight distortion arising from inequalities of temperature.

Glands of the labyrinth type are fitted at each end of each turbine, with pockets suitably arranged from which connections are taken to other parts of the system to control the glands; the outer pockets are kept at a pressure

slightly above atmospheric, being connected up to a common reservoir with a simple control from the starting platform. The gland where the shaft passes through the diaphragm separating the high pressure astern from the ahead turbine in the same casing, is also of the labyrinth type.

In view of the high temperature of the steam, monel metal is used for the blading of the high pressure ahead turbine and the first and second intermediate pressure turbines on each side. The astern impulse blades are also of monel metal.

Cast steel is used for all parts of the casing subject to high temperature, viz., the cylinder casings of the high pressure ahead, the first and second intermediate pressure turbines and the nozzle boxes.

CONDENSERS.

There are two condensers on each side of the vessel connected to the turbines by sub-divided exhaust branches. These latter are fitted with flap valves so that any one of the four condensers can be isolated from the system and opened out for examination in the event of failure of the condenser tubes. Each condenser can also be isolated on the water side, so that, if necessary, a repair can be executed without stopping the machinery. This feature is clearly illustrated in Figure 2, which shows details of the flap valves.

The object is to safeguard the system against entry of salt into the water-tube boilers. Salt detectors are fitted to each condenser to give a continuous indication of the degree of salinity of the condensate, the indicator being placed in a prominent position over the starting platform. An additional detector is also fitted in the feed pump discharge connections to the boilers.

All high pressure steam pipes are of solid drawn steel, tested to three times their working pressure. The flanges are screwed on to the ends of the pipes with a vanishing thread and the end of the thread at the face of the flange sealed by electric welding. The joint is made metal to metal with bolts of large diameter having a plus thread. Such joints have been tested at the works of the main contractor under severe conditions as regards temperature and pressure. Expansion of the steam pipes is provided for by a liberal use of bends.

REDUCTION GEARING.

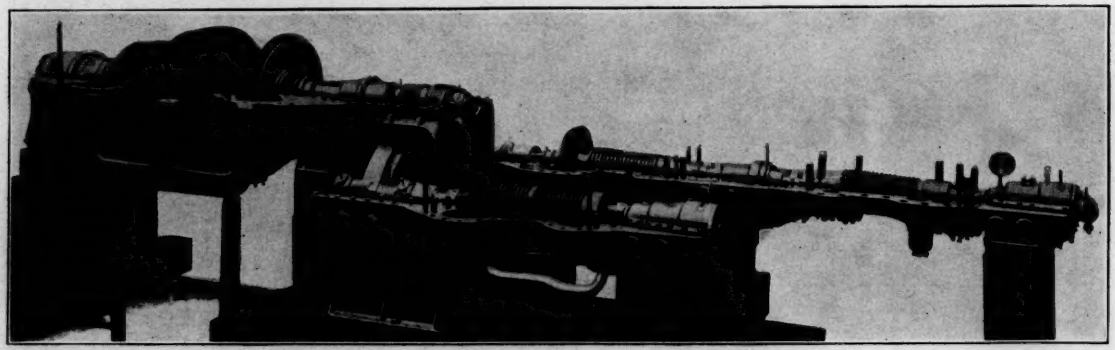
The gearing is of the usual double helical type, and consists of a wheel 54 inches diameter on each shaft with three pinions to each wheel. Those connected to the high pressure and intermediate pressure turbines are 5.13-inch pitch circle diameter, and those to the low pressure turbine 10.26-inch pitch circle diameter.

BOILERS.

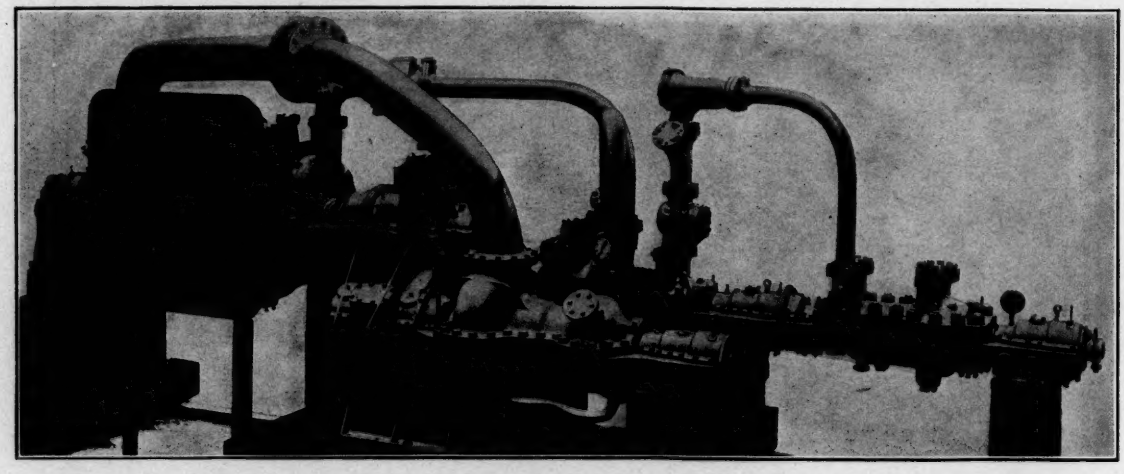
The boilers supplied by Messrs. Yarrow & Company, are two in number of the Yarrow watertube type, designed for a working pressure of 550 pounds per square inch. They are fitted with superheater elements to raise the temperature of the steam to a maximum temperature of 750 degrees F., and air heaters in the uptakes to heat the air on its way to the grates. They are fitted with grates for coal burning with a closed stokehold system of forced draft.

Means are provided for quickly reducing the fires when drawing near to the piers, by stopping the fan and shutting off the supply of air to the grates.

These boilers were described by Mr. Harold Yarrow in a paper read before the Institution of Naval Architects in March of this year.



PORT TURBINES AND REDUCTION GEARING WITH UPPER CASINGS REMOVED.
FIGURE 3.



PORT TURBINES AND REDUCTION GEARING COMPLETE.
FIGURE 3.

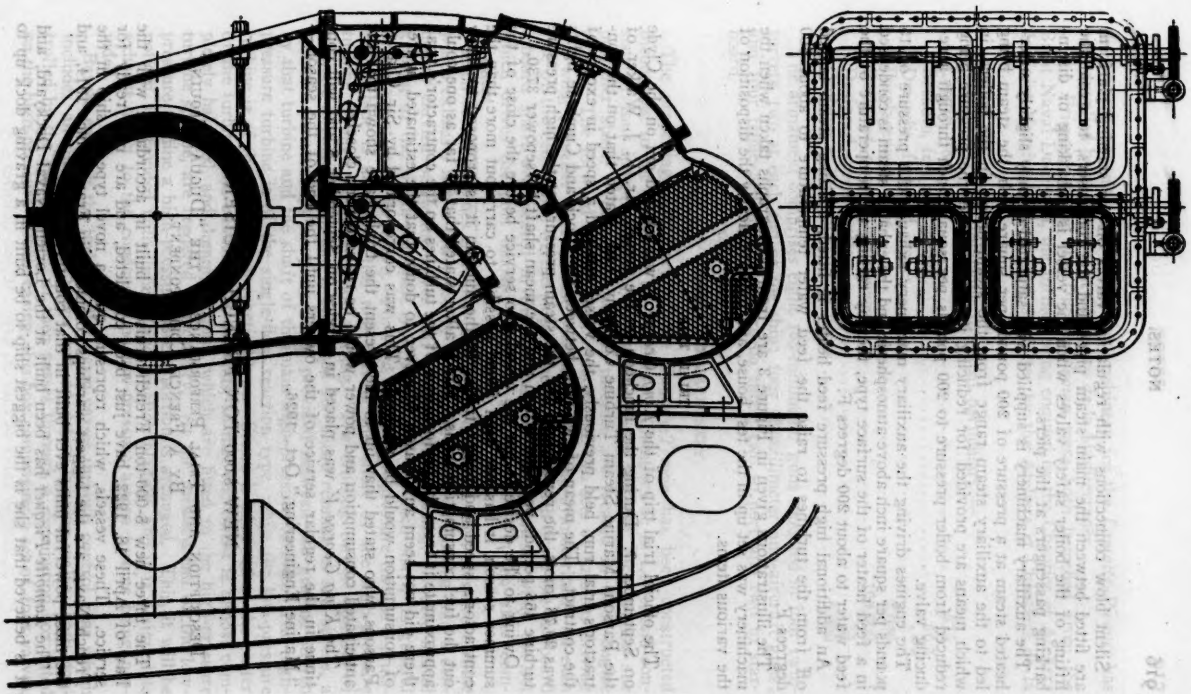


FIGURE 2.—ARRANGEMENT OF CONDENSERS, SHOWING FLAP VALVES.

Silent blow connections with regulating valves on the starting platform are fitted between the main steam pipes and the condensers, to avoid the lifting of the boiler safety valves when the vessel is embarking or disembarking passengers at the piers.

The auxiliary machinery is supplied with saturated or only slightly superheated steam at a pressure of 200 pounds per square inch, the steam being led to the auxiliary steam range from the steam drums in the boilers, in which means are provided for reducing its temperature, the pressure being reduced from boiler pressure to 200 pounds per square inch through a reducing valve.

The engines driving the auxiliary machinery exhaust at a pressure of 15 pounds per square inch above atmospheric and the exhaust steam is condensed in a feed heater of the surface type, thereby raising the temperature of the feed water to about 200 degrees F.

An additional high pressure feed heater is fitted, utilizing steam tapped off from the turbines to raise the feed water temperature to about 300 degrees F.

The illustrations given in Figure 3 are from photographs taken when the machinery was set up in the test house, and indicate clearly the disposition of the various items.

TRIAL TRIP.

The official trial trip of the *King George V* was carried out on the Clyde on September 7. During the luncheon on the vessel, Mr. R. J. Walker of the Parsons Marine Steam Turbine Company, Ltd., stated that on the contractor's trial run, held previously, the vessel attained a speed in excess of the contract. The mean of two runs between the Cloch and Cumbrae lights was 20.78 knots, the revolutions 582, the steam pressure at the high pressure turbine 465 pounds per square inch and the mean shaft horsepower 3730.

Owing to the desire to put the vessel into service before the close of the summer season, it has not been found possible to carry out more than the contractor's speed and power trials; in the autumn it is proposed to carry out the fuel consumption trials. Mr. Walker stated that so far as one could approximate it from the performance of the turbines on the contractor's trial, there did not seem any reason whatever to doubt that the estimated figures of consumption would be attained. This was corroborated by Sir Charles Parsons, who stated that up to the present the trials have shown that the anticipated consumption and power would be realized very closely.

The *King George V* was placed in service on September 8 and will continue in the regular service of the owners until the end of the season.—"Marine Engineering," Oct., 1926.

NEW 8,000 TON FRENCH CRUISERS.

DESCRIPTION AND TRIAL PERFORMANCE OF THE "DUGUAY-TROUIN."

BY A FRENCH CORRESPONDENT.

The three new 8-000-ton French cruisers, built in accordance with the Law of April 18, 1922, have just been completed, and are now ready for service. These vessels, which represent quite a novel type of ship in the French Navy, are the biggest warships completed in France since 1914, and the most powerful units ever built in that country.

The *Lamotte-Picquet* has been built at the Lorient Naval Dockyard, and it is believed that she is the biggest ship to be built in a graving dock up to

the present moment. The propelling machinery for the *Lamotte-Picquet* has been supplied by the Forges et Chantiers de la Méditerranée.

The *Primauquet* and the *Duguay-Trouin* were both launched from the Brest Naval Dockyard, the former being engined by the Chantier et Ateliers de Saint-Nazaire-Penhoet, and the latter by the Ateliers et Chantiers de la Loire.

The ships have the following principal particulars:—

Length, feet	575
Breadth, feet and inches	56.5
Draught, feet and inches	17.5
Displacement, tons	8,000
Designed power, horsepower	100,000
Designed speed, knots	34
Cruising radius, miles at 15 knots	5,000

The armament comprises eight 155-millimeter guns mounted in four twin-turrets, two at each end of the ship; four 75-millimeter anti-aircraft guns; and four triple 550-millimeter torpedo tubes, two to port and two to starboard.

PROPELLING MACHINERY.

The propelling machinery of the *Duguay-Trouin* has been constructed throughout in the works of the Ateliers et Chantiers de la Loire, and consists of four independent sets of Parsons steam turbines, each driving one propeller shaft. These four sets of turbines are located in three engine rooms. The forward engine room contains the two sets of turbines driving the wing propeller shafts, these sets being arranged symmetrically on each side of the center line of the ship. The turbines which drive the starboard inboard shaft are located in the intermediate compartment, while the set which drives the port inboard propeller shaft is fitted in the after compartment.

Each set of turbines includes, for ahead running, a high-pressure turbine which exhausts into a low-pressure casing, the latter exhausting into a condenser carried below the lower half of the turbine casing. Double helical gearing of the single-reduction type is provided for each shaft, its general arrangement being as shown. Forward of the high-pressure turbine a cruising turbine is arranged, this unit driving the shaft of the high-pressure turbine through single-reduction gearing. The cruising turbine consists of a single impulse wheel, and is only used for ahead running at low speeds, a suitable disconnecting coupling being provided between the gearing driven by that turbine and the shaft of the main high-pressure turbine. The high-pressure turbine is of the single-flow reaction type, while the low-pressure turbine is of the double-flow reaction type. Steam is admitted to the center of the turbine, and exhausted at the ends, so that end thrust is almost completely balanced, and dummy pistons, which are fitted to the high-pressure turbine spindle, are rendered unnecessary. At the forward end of the low-pressure turbine a high-pressure astern portion is arranged. This, it will be noticed, is provided with impulse and reaction blading, and it is designed to exhaust direct to the condenser, no low-pressure astern turbine being provided.

Before entering the turbines, the steam passes through control gear which includes an ahead-running valve and an astern-running valve, both controlled by the same wheel. It then passes through a distribution box fitted on the high-pressure turbine casing, this box comprising a set of four valves,

also controlled by the same hand wheel. The first valve controls the inlet of steam into the cruising turbine, corresponding to 1/11th of the maximum power, and to a speed of about 16 knots; the second valve controls the steam inlet to the first high-pressure expansion stage (corresponding to one-fifth of the maximum power); the third valve controls the steam inlet to the third high-pressure expansion stage (corresponding to one-half of the maximum power); and the fourth valve controls the steam inlet to the fifth expansion stage of the high-pressure turbine (corresponding to the maximum power).

For astern running, steam comes from the control gear and is led directly to the low-pressure turbine. There are two directly-controlled valves fitted in the inlet pipe to the astern nozzles which enable steam to be admitted to a variable number of nozzles, according to the number of boilers working and hence to the quantity of steam available. The wheel of the control gear and that of the distribution box are fitted on a control board which also includes all control gear for the hand regulation of the throttle valve, and of the discharge valve to the condenser.

Each condenser is equipped with three air ejectors of the Maurice Leblanc two-stage type. Each condenser is provided with two air pumps; driven by geared turbines, each of these pumps being sufficient to ensure the maintenance of a sufficiently high vacuum in the condenser for low speed steaming. The circulating pump for each turbine is also driven by a geared steam turbine.

Steam is supplied to the turbines by eight oil-fired watertube boilers of the small-tube type, working at a pressure of 18 kilogrammes per square centimeter (256 pounds per square inch). Each boiler has a heating surface of 1100 square meters (11,840 square feet).

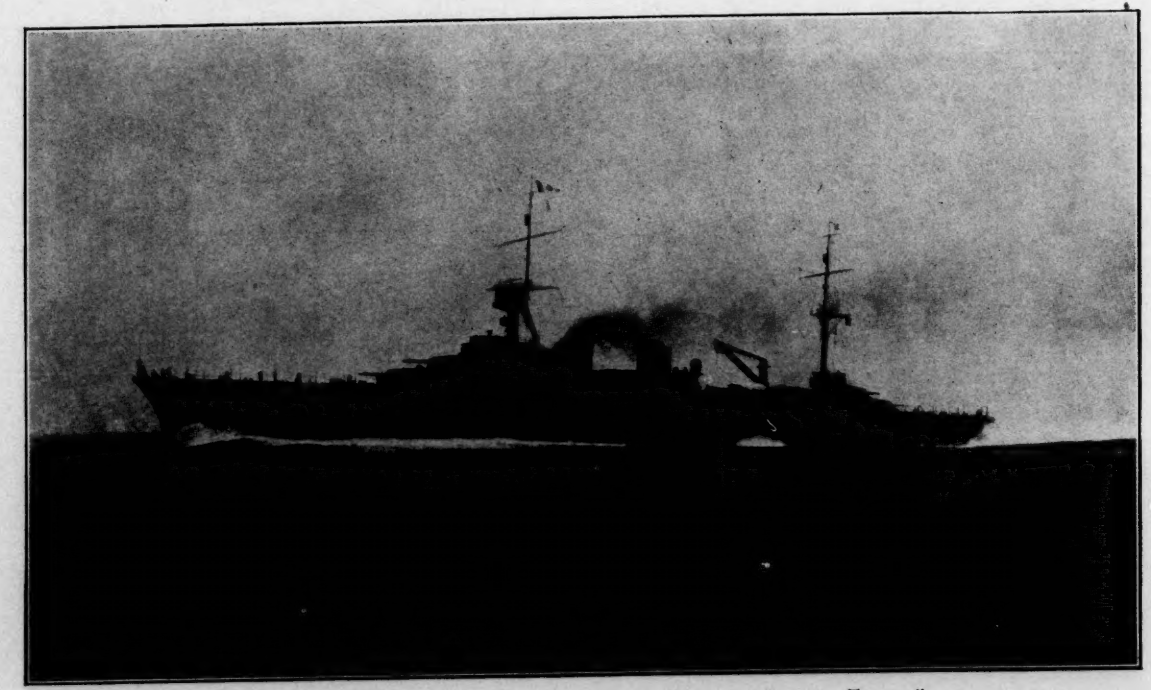
TRIAL PERFORMANCE.

The sea trials of the three cruisers included a 6-hour run at full power, a 24-hour resistance and consumption trial at 30 knots, and an 8-hour consumption trial at 15 knots.

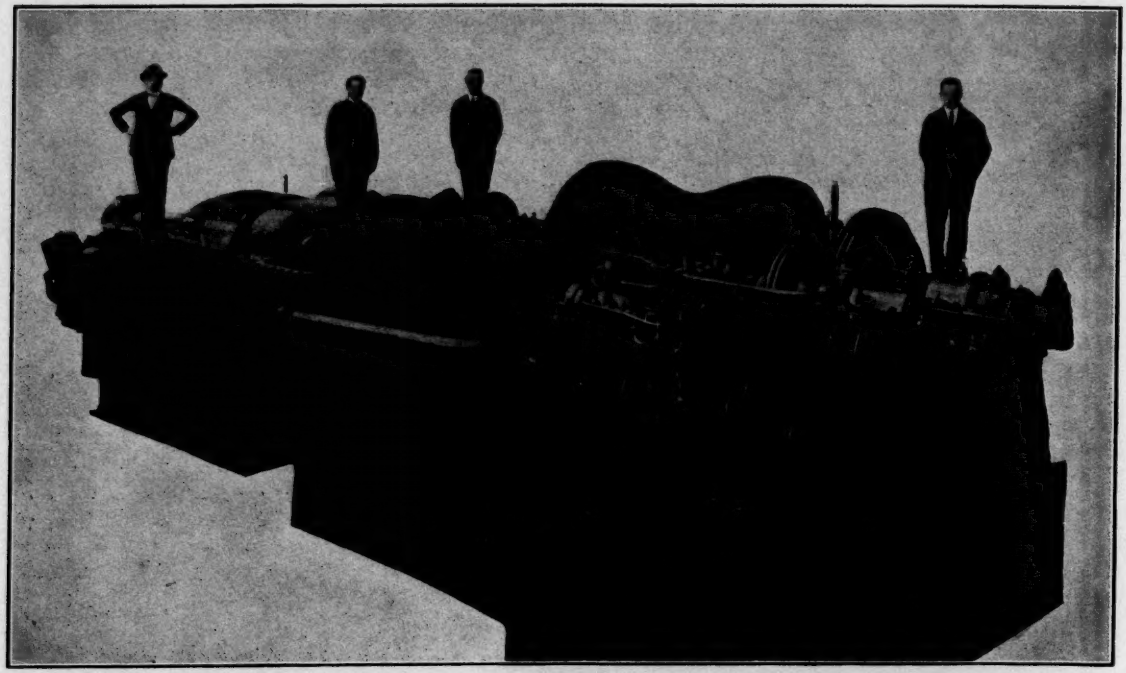
The following are the results of these trials for each vessel, from which

	Duguay-Trouin.	*Lamotte-Picquet.*	*Primauguet.*
Power trial:			
Speed on the measured mile, knots	33.60	33.04	33.066
Mean power developed during 6 hours, shaft horsepower	116,235	115,100	116,849
Mean fuel consumption per hour, kilogrammes	46,367	49,000	50,584
Resistance and consumption trial at 30 knots, during 24 hours:			
Consumption per mile, kilogrammes	804	820	825.44
Consumption trial at 15 knots, during 8 hours:			
Consumption per mile, kilogrammes	219.28	260	284.04
Distance necessary to stop ship when running at 20 knots, meters	425	600	550

(Calorific value of fuel oil used: 10,500 Calories.)



THE NEW 8,000-TON DISPLACEMENT FRENCH CRUISER "DUGUAY-TROUIN" STEAMING AT HIGH SPEED DURING HER TRIALS.



ONE SET OF PARSONS TURBINES AND GEARING, OF 27,500 S.H.P., FOR THE NEW FRENCH CRUISER "DUGUAY-TROUIN." NOTE CRUISING TURBINE GEARED TO FORWARD END OF H.P. TURBINE SPINDLE, AND "DOUBLE FLOW" LOW-PRESSURE TURBINE. HIGH- AND LOW-PRESSURE BLADING IS OF REACTION TYPE THROUGHOUT.

it will be gathered that while attaining a maximum speed of over 33 knots, these ships are capable of making continuous runs at 30 knots with a fuel consumption rate of only slightly over 800 kilogrammes (1766 pounds) per mile.—"The Marine Engineer and Motorship Builder," Oct., 1926.

NEW FRENCH FLOTILLA LEADERS.

Among the new warships ordered in accordance with the Law of April 18, 1922, which covered the first stage of the French post-war naval programme, were six flotilla leaders. Within a very short time these vessels will be in commission. The *Panthère* and the *Jaguar* have been built at the Lorient naval dockyard, the *Leopard* and the *Lynx* by the Atelier et Chantiers de la Loire, the *Chacal* by the Chantier et Ateliers de Saint-Nazaire-Penhoët, and the *Tigre* by the Ateliers et Chantiers de Bretagne, Nantes. The last-named is the first of the series to be delivered, and interest attaches to them in that these vessels represent an entirely new type of warship of which there is no precedent in the French Navy.

The order for the construction of the *Tigre* was placed with the Ateliers et Chantiers de Bretagne on February 26, 1923, and the first trials of the ship were performed in October, 1925, her delivery having taken place early this year. The following are the principal particulars of the *Tigre:*

Length overall, feet and inches	416.0
Length between perpendiculars, feet and inches	392.7
Maximum breadth at waterline, feet and inches	36.3
Depth, feet and inches	12.0
Displacement, tons	2,400
Designed power, shaft horsepower	50,000
Designed speed, knots	35.5
Cruising radius at 18 knots, miles	2,500

The armament includes six 5.1-inch (130-millimeter) guns, two 2.95-inch (75-millimeter) anti-aircraft guns, and six 21.65-inch (550-millimeter) torpedo tubes in triple mountings.

The ship is framed on the longitudinal system instead of the transverse system, as has hitherto been adopted for light French warships. The fine lines of the hull are a feature of the vessel, and it is noteworthy that a saving of 100 tons in weight was effected as compared to the estimated figures, which saving could have been utilized in increasing the armament or the amount of ammunition carried. The crew of the vessel is 8 officers, 33 non-commissioned officers and 165 men.

PROPELLING MACHINERY.

The propelling machinery consists of two sets of turbines, each set driving one propeller shaft through reduction gearing. There are also two cruising turbines, each driving one of the high-speed pinions. The turbines are of the Rateau-Chantiers de Bretagne type, and have been constructed at the Nantes works of the builders, who have devoted over twenty years to the development of the Rateau turbine to marine propulsion. The reduction gearing was supplied by Power Plant Co. Ltd., West Drayton. The condensers of the *Tigre* are of the Ginabat type, to which reference has previously been made in this journal, and Michell thrust blocks are fitted.

All auxiliaries are turbine-driven. The steam generating plant consists of five boilers of the Dutemple type, with small tubes. The boilers were constructed by the Ateliers et Chantiers de Bretagne after investigation at the naval works of Indret, and all are fitted with Cockburn safety valves. The oil-burning installation is of the Dutemple-Huge type.

The main auxiliaries in the three boiler rooms include:

2 reciprocating pumps and 2 heaters per boiler room for fuel oil.

2 centrifugal Weir feed pumps per boiler room, to feed boilers either directly or through the feed-water heaters (1 per boiler).

2 Rateau turbo-fans per boiler of a new improved type for forced draught.

TRIAL RESULTS.

The trials of the *Tigre* aroused considerable interest in France owing to the creditable performance achieved. The following trials were carried out:—8-hour speed trial, including measured mile runs, at a displacement of 2564 tons. The mean power developed during the eight hours was 51,843 shaft horsepower, the average speed being 35.92 knots. The total fuel consumption was 26.6 tons per hour, whereas the contract allowed a consumption of 28.3 tons per hour.

Following the 8-hour trial, the boilers were allowed one hour to work up to their maximum output, after which 57,200 shaft horsepower was recorded, the speed attained being 36.7 knots, and the fuel consumption 27.3 tons per hour.

A consumption trial at 18 knots recorded a fuel consumption of 311 pounds per mile only, being notably below the contract allowance. During the trials it was observed that when running at 18 knots the vessel could be stopped within two and one-half lengths. The change from 18 knots on the cruising turbines to running at the same speed with the main turbines was effected in less than two minutes. Turning trials were carried out at 20, 25, 30 and 34.5 knots, and showed that the diameter of the turning circle was between 500 and 600 m., *i.e.*, about four or five times the length of the ship.

A SPEED RECORD.

It is claimed that the speed of nearly 37 knots has never previously been attained by any warship of similar tonnage, and that an average speed of 35.9 knots for eight hours under full load conditions also constitutes a record.

Special stress has been laid in France on the important rôle that could be played in war time by such a ship as the *Tigre*. It is pointed out that in case of mobilization the vessel could convey over 600 soldiers from Marseilles to Algiers within 13 hours, so six vessels of that class could transport in a single night about 4000 men.—"Marine Engineer and Motorship Builder," Aug., 1926.

PROPELLING MACHINERY DEVELOPMENT.

The story of the development of ship propelling machinery is a fascinating one; and those who are interested enough to trace the line of progress from the earliest days of paddle-wheel ships and slow-speed, low-pressure steam engines will be amply repaid for their trouble. Whether one studies the subject deeply—tracing the introduction and eventual rejection of beam-engine and other types, novel and often unsuccessful valve gears, such radical departures from normal practice as Ericsson's "Caloric" engine, and other

features—or merely seeks to obtain a broad conception of early marine engineering history, one must be impressed by an outstanding fact, *viz.*, the steady rise of working steam pressures in the unending quest for improvement in machinery performance.

STEAM PRESSURES.

At the outset, steam pressures of a few pounds per square inch above atmospheric pressure were used in marine work. Marine engineers displayed a circumspection, which was perhaps wise, in the matter of higher steam pressures during the early stages of steamship development, and so progress was not rapid. Pressures gradually increased, however, and with progress in this direction came improvement in boiler design. Thus the story was continued, until eventually the compound engine and, later, the triple-expansion engine were forthcoming. With the introduction of this latter type of steam engine, some 52 years ago, steam pressures had reached a figure which was but 20 pounds per square inch or thereabouts below that used with the average triple-expansion engine of the present day.

Later in the century the multiple-expansion reciprocating steam engine installation was developed a stage further by the introduction of the quadruple-expansion engine, which enabled steam pressures to be increased still further and economy to be enhanced beyond the fairly satisfactory standards then existing. Earlier than this, some attempt had been made to obtain greater economy by the use of superheated steam; but for several reasons, chief of which were lubrication and metallurgical difficulties, practical considerations made its temporary rejection necessary.

PISTON SPEEDS.

With the steady increase in steam pressures, a concurrent rise in piston speeds was also effected. The significance of this statement can perhaps best be illustrated by a few representative design and performance data. Prior to the introduction of the triple-expansion engine, when two-cylinder compound-expansion machinery was extensively used in marine work, boiler pressures of from 40 to 60 pounds per square inch were common, and piston speeds in the neighborhood of 350 feet per minute were used. Such engines had a specific fuel consumption of about 2¼ to 2½ pounds of coal per indicated horsepower per hour.

About 1880 the average boiler pressure of marine installations had increased to about 80 pounds per square inch, and piston speeds of about 450 feet per minute were the order. The result of this progress, in terms of specific fuel consumption, was a reduction in the amount of coal burned per I.H.P. per hour to 2 pounds. With the fairly general adoption of Dr. Kirk's triple-expansion engine in the early eighties, steam pressures advanced to an average figure of about 160 pounds per square inch, and piston speeds as high as 550 to 600 feet per minute were considered good practice. This type of engine brought fuel consumptions down to 1¾ pounds per I.H.P. per hour, and in some cases the 1½-pound mark was even more closely approached. Further advance to 200 pounds per square inch at the boilers, with piston speeds of 900 feet per minute and over, came later; and with the fairly extensive adoption of quadruple-expansion engines, steam pressures some 20 pounds and more in excess of this figure were adopted, with beneficial results to machinery performance.

The result of these advances in marine steam engine practice was seen not only in more economical machinery, but—almost equally important to

the shipowner—in lighter and less bulky engines. This statement is really another way of saying that the economy of ships' propelling machinery was greatly enhanced, while the cost per indicated horsepower developed was reduced.

EARLY STEAM TURBINES.

Concurrent with the development of the quadruple-expansion engine was that of the steam turbine. Sir Charles A. Parsons took out his first patent for a steam turbine as long ago as 1884, but it was not until 1897 that he applied his invention to ship propulsion. In the intervening years the turbine had been improved in efficiency by adapting it for use with a condenser, thereby extracting the maximum amount of energy from the steam during its expansion. As everyone knows, the success of his first experimental vessel —the *Turbinia*, of 100 tons and 2100 shaft horsepower—led to the adoption of the new type of prime mover for the British destroyers *Cobra* and *Viper*, and, in 1901, for the Clyde pleasure steamship *King Edward*.

There is no need to dwell on the historical development of the marine steam turbine in the present outline, for the facts are too well known to warrant any recapitulation. It will be sufficient to remind readers that the successful development of the marine steam turbine was in the early stages, almost entirely due to the genius and tenacity of purpose of Sir Charles Parsons. It is almost superfluous, too, to remind shipbuilders that the steam turbine alone has made possible the existing high-powered, high-speed vessels of war and large Atlantic "greyhounds"; and even today, notwithstanding the popularity of the heavy-oil engine, it is probable that, were a very large and fast liner to be built, the type of propelling machinery chosen would be steam turbines.

The introduction of reduction gearing was an epoch-making development for which Sir Charles Parsons was also responsible. By interposing mechanical reduction gearing between the turbine spindle and propeller shaft, it has been possible to make use of higher turbine speeds, so that the ratio of turbine-blade speed to steam speed more closely approaches the theoretical ideal value for best efficiency. At the same time a gain in propeller performance is realized because of the lower propeller speeds, and consequent larger diameter of screws possible with this development.

In order to make the steam turbine suitable for the economical propulsion of slow-speed ships, mechanical gearing giving two stages of reduction—reduction ratios in excess of 50 to 1 are utilized in practice—was introduced a few years ago; and while the principle undoubtedly gave much superior fuel economy to that possible with the ordinary reciprocating steam engine, in some cases defects in the gearing manifested themselves in service, and the system received a setback. Trouble was generally confined to gears of the double-reduction type, and these were of such a nature that they were probably responsible to some extent for the intensive development of the marine heavy-oil engine during recent years. Later experience indicates, however, that the difficulties associated with double-reduction gearing have been overcome.

HEAVY-OIL ENGINES.

At this stage it may be convenient to outline briefly the development of the marine heavy-oil engine. This type of prime mover has evolved at a remarkable rate for cargo-ship propulsion during the past three or four years, and has shown itself to be an economical, reliable, and in every sense prac-

tically successful marine power unit. So great has its success been, in fact, that engine-builders have been emboldened to propose it for the propulsion of what are now regarded as moderately large passenger ships, with the result that several such vessels are now in service.

It is now some thirty-four years since the late Dr. Rudolf Diesel built his first high-compression internal-combustion engine—which, incidentally, used *coal dust* and not heavy oil as fuel—and it is twenty-nine years since the original licenses for the manufacture of the Diesel type of heavy-oil engine were granted to various firms. A further period of over ten years elapsed before this type of engine was applied to an ocean-going craft of any size. The success of such renowned motorships as the *Vulcanus* and *Selandia* demonstrated the suitability and durability of the heavy-oil engine for ocean work, and the development of this type of prime mover went steadily forward from about 1913. So far as shipowners in this country were concerned, however, they followed, for the most part, a somewhat conservative policy in connection with the serious adoption of the internal-combustion engine.

The intervention of the war necessitated the marine engine-builders of this country concentrating upon the production of standardized reciprocating steam engines and, later, turbine machinery for the propulsion of cargo-carrying ships, the great necessity for maximum output and absolute reliability naturally ruling out the further development of the heavy-oil engine. This condition of affairs allowed those foreign firms who had concentrated upon the marine oil engine prior to the war to intensively develop their designs during the war years, with the result that when British firms considered the construction of oil engines in their post-war programmes the majority of them followed the wise course of building to Continental designs, under license.

The post-war growth of the marine heavy-oil engine industry, both in this country and abroad, is too well known to justify further comment here. It may not be out of place, however, to touch upon the recent development of the double-acting type of Diesel engine, a forward step hastened by the demand for motor-driven passenger ships of fairly high power. Numerous double-acting designs are now in production or are in the experimental stage, and several passenger vessels have been, or are being, equipped with propelling machinery of this type.

HIGH-PRESSURE TURBINES.

There is no doubt that the very high economical performance of the marine oil engine and the unquestionable general soundness of modern designs have led builders of marine steam engines to watch closely the development of this new prime mover—a state of affairs which is desirable for all concerned. Particularly has the threatened inroad of the oil engine upon that once unchallenged field of the steam engine—the propulsion of large passenger ships—placed the partisans of steam on their mettle; and the most direct reply they have given is the new high-pressure steam turbine which Sir Charles Parsons has applied to the Clyde passenger steamship *King George V.*, a full description of which will be found elsewhere in the present issue. We believe that this vessel marks a new era in the history of marine propulsion, just as the *Turbinia* and the *King Edward* did in their day.

Sir Charles has stated that this new development should give an improvement in efficiency over the equivalent moderate steam-pressure turbine installation of from 10 to 15 per cent; and if this result is realized, as is un-

doubtedly more than probable, an advance of the utmost importance will have been made. If a 15 per cent improvement be assumed for the new type of turbine, the actual amount of fuel used per shaft horsepower per hour will be rather more than that of equivalent Diesel machinery, but against this must be ranged the reduced lubricating-oil bill, less weight, and lower cost of the steam plant. Sir John H. Biles' investigations of the comparative commercial efficiency of both passenger and cargo ships propelled by these special high-pressure turbines and by heavy-oil engines are carefully considered, thorough, and unbiased examinations of this vital question; and while their absolute value may not be great on account of the numerous assumptions necessarily made in his various calculations, his work has served to show that the motorship's superiority for various trades and routes is to be seriously challenged by this new steam-engine development.

This challenge to the marine oil engine—to say nothing of that of the poppet-valve reciprocating steam engine, from which very good results have been obtained in cargo steamships—is a healthy sign, which augers well for the future of shipbuilding and shipping. There is no doubt that both the steam and internal-combustion types of propelling machinery are capable of further development, and the immediate future promises technical advance of an extremely interesting nature on both sides.—"The Shipbuilder," Oct., 1926.

PULVERIZED FUEL FOR MARINE BOILERS.

At different times the value of powdered fuel for use under marine boilers has been commented upon in these pages. As is generally known, pulverized coal firing has been successfully applied to land boilers for a number of years, particularly in America, but up to the present its application to marine boilers has been considered to entail certain difficulties of design and operation which have not proved easy to solve. At the present time, for large power stations the claims of pulverized fuel firing and mechanical stoking are equally strong; and it may be said that both systems of firing have reached an equally high degree of efficiency.

Mechanical stokers have been applied in one or two instances to water-tube-boilered steamers and have given excellent results. It will be recalled that an account of experiences with two small steamers fitted with mechanical stokers was given in a paper before the Institution of Naval Architects by Mr. W. J. Muller in 1923. The overall boiler efficiencies obtained were excellent, and the general operation of the boilers and stokers was apparently satisfactory, for the same company has lately ordered four small passenger and cargo vessels which will be equipped with mechanically stoked water-tube boilers.

DIFFICULTIES TO OVERCOME.

In view of the fact that mechanical stoking can successfully be applied to marine boilers, the question naturally arises: why cannot the apparently more desirable method of burning coal, namely, by finely powdering it and injecting it into the boiler furnaces in much the same manner as oil is burned, be applied to marine boilers. It is generally thought that the extreme length of flame usually associated with pulverized fuel firing, and the necessarily large combustion chamber volume required for the efficient combustion of powdered fuel, militates against its practical adoption on shipboard. Recognizing these facts, experimental work has been carried out in America with a type of burner for powdered coal firing intended to give efficient combustion with a reasonably short length of flame, and hence

with boilers of moderate overall dimensions. It is understood that these experiments have been of an encouraging nature and probably more will be heard of the development in the near future.

Engineers in this country, on the other hand, have tackled the problem from a different standpoint and have proposed to adapt existing pulverized coal practice to marine use by attaching an external combustion chamber to the front of what is substantially a normal water-tube boiler, such as would be used for oil fuel firing. In this way the difficulty associated with the characteristically long flame of powdered coal would be overcome and the total combustion chamber volume would be sufficiently large to ensure thorough mixing of the furnace gases and, hence, good combustion. It would appear that such a course is sound in every way, because it is a well-known fact that powdered coal has the same essential characteristics as liquid fuel.

As illustrating this last remark, it might be recalled that both powdered fuel and boiler oil are fluids; both may be syphoned, pumped or ejected; both are under positive control as regards quantity conveyed; both burn better when intimately mixed with oxygen supplied by heated air; both, from the time of entry into the furnaces, may be termed gaseous fuels, and require more combustion space than raw coal; and both, if the coal is bunkered in the powdered state, require the same labor conditions in burning. From the foregoing remarks it would appear reasonable to hope that the efficient utilization of pulverized fuel under the boilers of mercantile and naval vessels will be an accomplished fact in the near future. The majority of mercantile steam-driven vessels, however, are fitted with cylindrical or Scotch boilers, and the view is generally held that powdered coal firing of such steam generators is not practicable at the present stage of development. The limited combustion chamber volume, the short path which the flames would traverse before striking the boiler tubes, and the possibility of trouble arising from the accumulation of fine dust in the boilers, are the objections usually cited against the employment of this type of coal-burning under marine Scotch boilers.

PRACTICAL RESULTS WITH SCOTCH BOILERS.

It will, therefore, be of more than ordinary interest to many marine engineers to recall the successful practical trials which were carried out four years ago in Australia on a normal single-ended Scotch boiler fitted up for powdered coal firing. As in the case of the proposals put forward in this country for the application of pulverized fuel firing to marine water-tube boilers, an external combustion chamber was attached to the front of the boilers upon which the experiments were carried out. These experiments were conducted by Engineer-Cdr. J. J. C. Brand, of the Royal Australian Navy, and the modifications to the boiler and the powdered coal apparatus were also designed by the experimenter. An account of these experiments, and other work on the subject, was given in a paper which Engr.-Cdr. Brand read before the Institute of Engineers in Australia in June, 1923. The notes given hereunder in connection with Cdr. Brand's work have been prepared after a study of his very interesting paper, and we take this opportunity of acknowledging our indebtedness to the author of the paper and the Institution of Engineers in Austrialia for the information obtained therefrom. The boiler upon which the experiments were carried out was a single-ended return-tube boiler 11 feet in diameter by 9 feet 3 inches long and fitted with the corrugated furnaces. The heating surface was 980 square feet and two combustion chamber volume 200 square feet. An external chamber is built on to the front of the boiler, the actual burner, air trunks, coal con-

veying and mixing gear being installed inside this casing. The main air supply was taken from a Howden air heater contained in the smoke-box of the boiler and the coal was fed to the ready-use bin along pipes located on the bunker floor. The best arrangement of extracting the coal-dust continuously from the main bunkers and delivering it to the ready-use bin was found to be along 4-inch pipes slotted on the bottom, the coal being drawn through these pipes and led to the warm air near the casings in the stokehold, and so to the bins. These pipes were each fitted with a shut-off cock operated from the lower stokehold platform. A 11-inch gas pressure pipe was introduced in the supply pipe in case of stoppage, but was never required during the experiments. The slots in the pipe were covered with a semi-circular sleeve working in guides, the sleeve being manipulated by means of a bell crank and screw spindle, which passed through a stuffing-box in the bulkhead and had a wheel and index plate in the stokehold. The 4-inch supply pipe was led through the lower part of the bunker bulkhead to a suction fan capable of maintaining an 8-inch vacuum. The fan was situated above the ready-use bin and discharged the coal dust to the latter through a cyclone separator. When it was desired to extract fuel from the bunker, the fan was started, with the cock to the stokehold opened, and the sleeve rotated until the ports were open the desired amount, when fuel passed to the bin until the fan was stopped.

In the process of pulverization coal is aerated, and so behaves as a liquid; but during charging in the bunkers the vibration of the vessel de-aerates the fuel, and hence in the experiments carried out in Australia, fluffing pipes were introduced just above the bunker and in the ready-use bin, so as to maintain aeration by the injection of inert gas at a pressure slightly greater than that of the head of the powdered fuel. The occasional use of these fluffing pipes was found to maintain the fuel in a considerable liquid state. During the three months the boilers of this small vessel were operated with powdered coal as fuel, no appreciable wear and tear on the fan impeller and pipes was noticeable, a condition of affairs which one would anticipate in view of the very fine pulverization and graphite-like texture of efficiently pulverized coal. Two methods were used in the experiments to determine the amount of fuel in the ready-use bin. One consisted of a glass sounding tube having one end open to the fuel, the other end passing through a stuffing box in the top of the bin, and being fitted with a rubber valve. A second device was also fitted for the same purpose, a rectangular projection fitted with a protected triple-glass window running the depth of the bin and being illumined on one side by an electric lamp. The coal for the trials described was dried and pulverized ashore, the moisture content never exceeding 2 per cent, while pulverization was carried to the point when 85 per cent passed a 200 mesh, the remainder passing through a 100 mesh. The coal was transferred from the shore station to the vessel through 6-inch pipes capable of discharging 10 tons per minute. Inert funnel gas, under pressure, was passed along the bunkering pipe, the gas picking up the powdered coal as it passed over the surface of the fuel in the short storage bin. A pressure of 30 pounds per square inch behind this gas was sufficient to transfer the finely-divided dust to the air-tight storage bins aboard the vessel. A separator was fixed at the bunker inlet, so as to relieve the gas pressure in the bunker. By conveying the fuel to the bunkers in this novel way the particles of pulverized coal were enveloped in the inert gas, so that the CO_2 portion of the gas, being heavier than air, displaced the latter until the bunker was filled with powdered coal and a gas incapable of supporting combustion. In this manner the possibility of an explosion occurring in the bunker or the supply pipes to the ready-use bin was avoided. (It is interesting to note that pulverized

coal was on one occasion stored for six weeks in a bunker at a temperature of 125 degrees F. without any trouble arising.) The method of using inert gas as described above, and the burner apparatus, a description of which follows, have been patented.

The Brand burner receives its fuel from a bin, the outlet of the latter having a gate control, so operated that no dust can escape. To prevent arching or packing of the coal, an air pipe, controlled by a cock, supplies air just above the gate. The coal drops from the bin into a rope drag conveyor, which travels past all the furnaces and back to the bin. Opposite each furnace a section of the conveyor pipe contains the feeder box. In this a worm rotates, feeding the coal forward to a downcomer, where it is picked up by the primary air which has been warmed by a Howden heater.

To prevent packing, an air blast is led into the back end of the screw chamber. The worm is operated by a friction drive, rotated by a shaft driven by the gear which works the drag conveyor. The feed can consequently be varied at will from minimum to maximum or, by depressing the handle of the driven disc, it can be stopped instantly. The primary air supply is controlled by a butterfly valve at each furnace.

MIXING THE FUEL.

The mixing of the fuel is effected in a novel manner. Below the entrance point of the fuel the pipe is jointed and coned out from both sides of the joint. A diaphragm of 80-mesh gauze wire in a carrier is placed at the joint, and the passage of fuel and air through this causes intimate mixing. The diaphragm is inserted so that it can be withdrawn and a clean one substituted in a few seconds. Just before leaving the mixing pipe the mixture is given a rotary motion by a turn and a half of helical vane. The end of the mixing pipe is covered by the burner pipe which slips over the former, and can be adjusted by means of the handles to ensure that the zone of conflagration begins just at the point of admission of the secondary air. When lighting up the pipe should be forward; it can afterwards be drawn back as the brickwork becomes incandescent. Secondary air from either a closed stokehold or a Howden heater is introduced into an air box controlled by flaps. This air passes through a ring containing vanes curved in the opposite direction to those in the burner pipe. This causes an eddy or vortex at the entrance to the combustion chamber, prevents stratification of the gases, and ensures that the fixed carbon is consumed in the shortest possible distance from the inlet.

LOW TEMPERATURE DISTILLATION AND POWDERED FUEL.

Before, during, and since the time that the trials on pulverized coal were carried out in Australia, a method of low temperature distillation, known as the "L. & N." process, was being developed by the Sensible Heat Distillation Co. Ltd., of 100 Victoria Street, London, S.W.1, whereby oils, and gas for power and town use, are distilled from coal, leaving a solid residue ("L. & N." fuel), which has been proved to be more suitable for burning under boilers than coal.

The following is a short description of the method of distillation which results in the production of gas, "L. & N." oil, and "L. & N." fuel, from coal, and in the first place it should be noted that small coal, which is relatively valueless, is just as good for distillation purposes as the larger coal for which good prices can be obtained.

Coal is fed into one end of a slowly rotating, inclined, and firebrick-lined retort. The inclination of the retort, and the speed of its revolution, cause

the coal to travel along its length at a pre-determined rate. It is met by a stream of hot, inert gas, which is admitted at the other end. This gas sets free the liquid and the gaseous matter and passes one to the coal end of the retort, giving up some of its heat on its journey, thus gradually heating the coal, and cooling itself at the same time, while retaining the oil in suspension. Due to this counterflow cycle, the oils never get heated above the temperatures at which they are evolved, and there is no danger therefore of their cracking; the oils remain as primary oils, and the lubricating fractions distilled from them are actually of a higher lubricating quality than proprietary brands against which they have been tested. They retain their lubricating qualities at both higher and lower temperatures than a great number of oils obtained from natural well oil. The coal, in being transformed into "L. & N." solid fuel, retains its cellular structure, and the cells, due to the counterflow method of distillation, do not get crated with graphitic matter, and this leads to easier pulverization and greater combustibility. During the preparation of the "L. & N." solid fuel, a certain amount of small material is formed; this can be conveniently burned, to a certain extent, in boilers or furnaces situated near the distillation retorts. A large scale plant, capable of distilling 10 tons of coal per day, is in operation at Barnsley, Yorkshire, and some interesting tests were recently carried out on "L. & N." fuel in a Woodeson type water-tube boiler designed for pulverized coal firing at the Victoria Works of Clarke, Chapman & Co. Ltd., Gateshead. The boiler was fitted with a Woodeson turbo-pulverizer and Clarke, Chapman & Co.'s own type of burner. We had an opportunity of witnessing these trials, which were completed in a very satisfactory manner, and below we give some particulars of the boiler used and the results obtained.

Particulars of the Boiler.

Heating surface, square feet	5,200
Working pressure, pounds per square inch	250
Combustion chamber, cubic feet	3,400
Length of passage of flame before striking tubes, feet	34
Burners—	
Number of	4
Area, each, square inches	21
Capacity, each, pounds per hour	1,000

Particulars of the "L. & N." Fuel were:—

		Per cent.
Composition—	Moisture	3.34
	Volatile	4.60
	Fixed carbon	73.20
	Ash	18.86
		100.00
As received—	Above 1 inch	60.00
	½ inch to 1 inch	19.70
	Below ½ inch	20.30
		100.00
As pulverized—	Through 100 mesh	95 to 98 per cent.
	Through 180 mesh	80 to 85 per cent.
		Cubic feet per ton.
Aerated or "fluffed," volumetric gravity		0.61, or 57.5
After 18 hours' settlement, volumetric gravity		0.664, or 52.2
After 36 hours' settlement, volumetric gravity		0.670, or 52.3
De-aerated by vibration volumetric gravity		0.91, or 38.5

(N. B.—Bituminous coal after pulverization and 36 hours' settlement averages 46.6 cubic feet per ton.)

These figures are of great interest, as they indicate that a lighter draught is required in air-swept separators or pulverizers for residue from distillation than is required for pulverized coal.

Results of the Trials were—

Feed-water temperature, degrees F	112
Boiler pressure, pounds	218
Superheat, degrees F	537

Ash—Fine white powder containing no combustible matter.
65 per cent in bottom of chamber.
30 per cent in soot hoppers.
5 per cent to atmosphere.

	Per cent.
CO_2 in flue gases, averaged	17
Power for grinding compared with that required for coal	90
Power for grinding compared with that required for gas coke	85
Water evaporated from and at 212 degrees F., pounds	69,300
Water evaporated per pound of fuel, pounds	9.2
Thermal efficiency, per cent	83.5

Flame—Noticeably pure and white.

As showing the striking economies which can be derived from the use of pulverized fuel on board a steamer fitted with high-pressure turbine machinery, we have taken the figures which were contained in Table I of Sir John H. Biles' I.N.A. paper on "The Relative Commercial Efficiency of Steam Turbine and Diesel Machinery for Cargo Vessels," and have added another column showing the annual operating expenditure which can be anticipated were "L. & N." residue pulverized fuel used instead of raw coal or oil. The cost of pulverizing this fuel and shipping it in the powdered state have been taken into account in preparing the figures put forward, and the net cost of such fuel per ton has been taken at the conservative figure of 20s. The comparative figures are given in the table below:—

S.H.P., 2,700.	H.P.T. with "L. & N." pulverized Residue Fuel.	H.P.T. with Coal.	H.P.T. with Oil	Diesel.
First cost of machinery	£45,000	£45,000	£45,000	£63,000
Weight of machinery, tons	480	480	480	670
Fuel—				
Per S.H.P. per hour, pounds	0.88	1.0	0.67	0.42
	Tons.	Tons.	Tons.	Tons.
Per day at sea	25.45	28.93	19.28	12.15
Per day in port	5.0	5.0	3.0	3.0
Per year at sea, 250 days	6,362.0	7,233	4,820	3,038
Per year in port, 115 days	575.0	575	345	345
Total fuel per year	6,937.0	7,808	5,165	3,383

	20s.	23s.	65s.	80s.
Cost of fuel per ton	£	£	£	£
Cost of fuel per year	6,937	8,979	16,786	13,532
Cost of lubricating oil per year	140	140	140	825
Engine-room wages and food	2,950	3,800	2,950	2,750
Fresh water	107	107	107	16
Charges, 18½ per cent of cost	8,325	8,325	8,325	11,655
Total	18,459	21,351	28,308	28,778

—"The Marine Engineer and Motorship Builder," Aug., 1926.

HIGH EFFICIENCY STEAM INSTALLATIONS FOR SHIP PROPULSION WITH SPECIAL REFERENCE TO THE QUESTION OF AUXILIARY MACHINERY.

By Stanley S. Cook.

With improvements that have been and are being made in the consumption of the main engines, the auxiliary machinery is now responsible for a considerable percentage addition to the fuel consumption. In this paper the consumption of auxiliary machinery is discussed and analyzed and suggestions are made as to the most economical arrangement, making use of regenerative feed heating, and comparative figures worked out for the total fuel consumption attainable with a high efficiency design of main turbines for a cargo vessel of 5000 shaft horsepower with the auxiliaries (1) steam-driven, (2) steam-electric, (3) Diesel-electric, and (4) mechanically-driven from main engine.

The estimated oil consumptions per shaft horsepower hour for all purposes for the four cases considered are (1) 0.667, (2) 0.648, (3) and (4) 0.62 pound.

It is noteworthy that most of the papers which have been devoted in recent years to subjects relating to power production, whether for service on land or for the propulsion of vessels, have had for their keynote the question of thermal efficiency, which indicates that at the present time every avenue is being explored which promises to lead to improvement of efficiency.

Developments in land installations of steam turbines for the generation of electric power have brought into prominence the value of securing, by improvement in practice, a conversion of heat into mechanical energy on the highest scale of efficiency that thermodynamic laws allow. Reference need only be made to the paper by the Hon. Sir Charles A. Parsons at the World Power Conference last year, in which are given the thermal efficiencies attainable in steam plants under practical conditions by the use of high boiler pressures and temperatures, with cycles of operation following known thermodynamic principles. In this paper it is shown that with such steam plant thermal efficiencies from fuel to electricity can be realized equal to 30 per cent, and fuel consumption as low as 0.45 pound of oil per brake horsepower hour. In that paper an actual installation is described of a turbine plant of 50,000 kilowatts capacity in which results in the neighborhood of these figures are expected to be obtained.

Such results are made possible with maximum temperatures within safe values for present-day materials, by using known practical means of increasing the available energy per pound of steam.

A review of the history of the steam turbine in marine propulsion discloses the fact that during the past 20 years there has been an increase in overall efficiency of practically 100 per cent made up of improvements in turbine efficiency and propeller efficiency principally through the introduction of gearing between turbine and propeller, and an increase of available energy due to the use of superheat.

The use of mechanical gearing has now enabled turbines in marine propulsion to be designed with the maximum efficiency of conversion of the available energy of the steam as on land, and following the lead of land turbines it is now being sought to realize the considerable improvement which is attainable on further increasing the available energy per pound of steam, by the use of higher steam pressures and temperatures and by regenerative heating of the feed water by previously expanded steam.

Many concrete designs for such installations have been worked out. Two examples for total outputs of 27,000 and 13,500 shaft horsepower were described in a paper read by Sir John Biles at the meetings of the Institution of Naval Architects this year, and comparisons made on an economic basis with installations of other types for the same vessel, principally with Diesel engines, to the advantage of the high efficiency steam turbine plant.

The most efficient arrangement of marine turbines in recent installations is recognized to be one consisting of three or more turbines in series grouped around a common gear wheel through which their power is transmitted to the propeller shaft. With this type of installation, the turbines run at a high speed of rotation and a high relative blade speed, and at the same time are of moderate size, easily overhauled and of rigid construction. Such an arrangement has therefore been retained for the new designs with their greater range of expansion.

Since economy in total fuel consumption is the ultimate aim, while the highest efficiency of conversion of heat into power is necessary on the part of the main engines that propel the vessel, equal consideration must be given to the question of auxiliary machinery which, with the improvements that have been made to reduce the consumption of the main machinery, is now responsible in a complete installation for a very appreciable percentage of the total fuel consumption. Wasteful auxiliary machinery either through unsatisfactory provision in the first place or through want of care in its maintenance and management has not infrequently in the past caused a poor overall performance, for which the propelling machinery was blamed.

Mr. Archibald Gilchrist has cited a case of excessive auxiliary consumption. He states, "In another vessel where lack of economy was apparent, after the introduction of water meters in order to localize the inefficiency, it was found that the auxiliaries were using 44 per cent of the total steam used by the main engines. After these auxiliaries were thoroughly overhauled and put in good working order, the consumption dropped to about 15 per cent of the steam used by the main engines."

The auxiliary machinery includes pumps for feeding the boilers, air and circulating pumps for the condensing plant, fans for the supply of air to the furnaces under forced draft, oil pumps for the lubrication of the bearings and gearing, and it is of vital importance that all of these should be entirely reliable in operation. Although the power required to drive the various pumps is small, their rate of consumption is usually high. In direct acting pumps for example the steam is used non-expansively, and a high efficiency cannot be expected. Figure 1 is a typical curve of consumptions per brake horsepower for such direct acting steam pumps.

The effect upon the total fuel consumption is not, however, obtained by simple addition of the auxiliary steam consumption to that of the main tur-

bines. It is generally the practice to exhaust the steam of the auxiliary engine against a back pressure a few pounds above atmospheric, which enables the heat of this steam to be re-utilized for feed heating. The feed heater in fact fulfills the role of a condenser for the auxiliary exhaust steam, with this difference that the cooling fluid in this case being the feed water itself the heat that it absorbs is not lost to the system.

In all modern installations as much as possible of this heat is recovered in the feed heater, and it is sometimes held, because the heat of the auxiliary

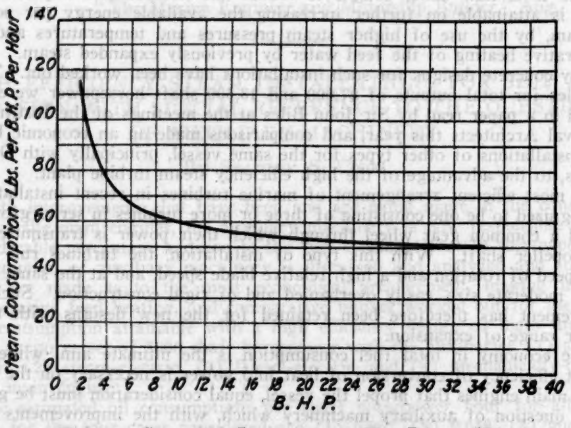


FIGURE 1.—TYPICAL CURVE OF CONSUMPTION PER BRAKE HORSEPOWER FOR DIRECT-ACTING STEAM PUMPS.

exhaust steam can be re-utilized in this way for heating the feed water, that the actual cost in fuel consumption for driving the auxiliary machinery is small. This, however, is not a true view of the case, since there are two other economical means of supplying the required low grade heat to the feed water.

The first of these has frequently been employed in land installations. Although it has no exclusive claim to the name, it has succeeded in being known as the economizer, which is presumably evidence that it was the first practicable means adopted for the purpose in question. In the economizer the feed water is heated by passing through a group of tubes situated in the uptake of the boilers over the outside of which the gases of combustion flow on their way to the chimney. Since the heat carried away in the chimney is to a great extent a measure of the inefficiency of the boilers, it is easiest to appreciate the advantage of the economizer by regarding the cold feed water as a means of cooling the chimney gases. The economizer is not used in marine installations, partly because of the large amount of heat that is usually available for the purpose of feed heating in the auxiliary exhaust steam, and partly on account of the large space that an economizer would require. Even in land installations it is now being recognized that a better means of cooling the chimney gases is to utilize their heat for pre-

heating the air supplied to the furnaces. Since the quantities and specific heats of the incoming air and the outgoing gases are practically the same, this arrangement can be made very nearly a truly regenerative process.

The other means of heating the feed water economically is by the use of steam tapped off from the turbines at suitable pressures, after having given up a certain amount of its energy in the form of work by expansion down to the temperature at which it is required, and that method would be available if there were no auxiliary exhaust steam. It will be realized that the use of auxiliary exhaust steam for feed heating is in essence an application of this principle, the difference being that on account of the inferior efficiency of the auxiliary machinery, for a given amount of heat supplied to the feed water, much less work is obtained from it beforehand than if it had been tapped off from the main turbines.

It will therefore be seen that it is by no means a simple matter to assess the true cost in fuel consumption of driving the auxiliary machinery.

While it is not the purpose of this paper to describe in detail the auxiliaries which are essential for the efficient operation of the propelling machinery and the service of the ship under voyage conditions, but merely to determine the addition which they make to the fuel consumption, it will assist towards this end to enumerate briefly the various components with attention to the conditions which bear upon the subject of this enquiry.

First in order of importance is the boiler feed pump. Although treated as an auxiliary it is an indispensable part of the heat engine, in that, along with the air pump, it completes the steam cycle by raising the condensed steam to boiler pressure. In a large installation it is operated under float control so as to keep the water in the boilers at a constant level. For the sake of simplicity of control and reliability of operation, the slow direct-acting vertical type of pump is usually preferred. The relative diameters of steam pistons and plunger are such that the pump can work easily against the boiler pressure. The water pressure and the steam pressure are therefore inter-related and excessive pressure in the pump barrel cannot under any circumstances be reached. Admission of steam to the steam cylinder continues nearly to the end of the stroke, so that this engine works practically non-expansively. The available energy per pound of steam is little more than half of what it would be with adiabatic expansion; and if we allow for condensation and mechanical losses the best result that can be expected with a steam pressure say of 200 pounds per square inch and back pressure 10 pounds gauge, is a consumption of about 45 pounds per brake horsepower. The exhaust steam from this pump is available for feed heating.

Next in order of importance is the air pump. Inasmuch as a portion of its work is to pump the condensate up to the feed tank, it is, as regards the condensate, also a part of the heat engine. It has also to remove the air from the condenser, which if a good vacuum is to be maintained in close correspondence with the vapor pressure of the condensate, can only be done by withdrawing along the air a large proportion of vapor. The volumetric capacity of the air pump is therefore usually many times that of the volume of condensate to be extracted, and the air pump regarded merely as a condensate pump has but a low efficiency. Some care is required at the hands of the engineers to maintain the air pump running at the best revolutions, high enough to maintain the vacuum, but avoiding excessive speed which would involve waste of steam.

In modern plants the extraction of air with its inevitable accompaniment of vapor is performed either by a vacuum augmentor working in series with the air pump, or by an independent set of ejectors in series, and in the latter case a rotary pump may be employed for the extraction of the condensate.

The steam engine operating the air pump is usually also of the direct-acting type, and involves a large rate of consumption per brake horsepower, but the brake horsepower is difficult to assess, since in this case it depends upon the volumetric capacity of the pump rather than upon the duty it has to perform.

The circulating pump has the simple duty of pumping a certain quantity of cooling water against a head, which depends chiefly upon the resistance of the condenser tubes and circulating pipes, and which in a modern cargo vessel is of the order of 15 feet. This pump is usually driven by a simple engine with about 65 per cent cut off, or by a compound engine. The latter is of course the more economical. If it is desired to utilize the exhaust steam for feed heating, it is allowed to exhaust against a back pressure, but where sufficient exhaust steam is available otherwise for feed heating, the circulating pump engine may be best arranged to exhaust to a vacuum. Care must be exercised by the engineer to avoid running this pump at too high a speed, since its power and therefore the steam consumed, will rise practically as the cube of the speed.

Fans are necessary for the supply of air to the boiler furnaces, under a slight pressure. These, too, may be driven by compound engines, exhausting either against a back pressure or to vacuum. Their duty in air horsepower is easily calculated from the quantity of air required for the furnaces, and the air pressure at the fan. It is important that circulating pumps and fans should be designed as nearly as possible for the conditions under which they are to be employed, that their output characteristic may be at the point of maximum efficiency.

Another indispensable accessory is the oil pump, for supplying lubrication to the bearings. Such pumps have to work against a considerable head, in order to be able to force the oil through the oil coolers and still leave a good pressure of supply to the various bearings. The head provided for is usually about 50 pounds per square inch, and for a geared turbine installation the quantity of oil required to be circulated is roughly speaking two gallons per hour per shaft horsepower. These also are direct-acting steam pumps, and being small they have a steam consumption not less than about 60 pounds per brake horsepower.

In an oil burning installation there is also required a pump for the supply of oil fuel to the burners at the requisite pressure, and in a coal burning ship an ash hoist or ash ejector. These require to be supplied with a small amount of steam.

A steering engine, electric light engine, bilge, sanitary and general service pumps working as required are necessary for the service of the ship, and the steam that has to be supplied is included in the auxiliary consumption. The amount of this additional steam can only be based on service experience.

A certain amount of steam is continually being lost to the system through leakage of various sorts, and this must be included in the total consumption. Further, this implies a corresponding loss of feed water, which must be replaced from the reserve fresh water or by the distillation of sea water. The latter involves an expenditure of steam from 1.25 times to twice the amount of water distilled, which expenditure must also be included in the total consumption. The steam thus used for evaporation is condensed in the evaporator coil and returned to the system as hot water, while the distillate can be condensed either separately or in the feed heater.

For reliable data as to the consumption of the most important auxiliaries, reference may be made to a paper by Mr. T. G. Potts, read before the North-East Coast Institution of Engineers and Shipbuilders in 1923, describing measurements which were made on sea trials, using both measuring

tanks and water meter for the main condensate, and a second smaller water meter for the condensate of the auxiliary steam, which was led partly to the surface feed heater and the surplus to the winch condenser, and all drains from heaters and from the winch condenser led to the auxiliary cascade filter tank. These measurements were made at varying output of the main engines, during the contractors' trials, and measurements continued with water meters during subsequent voyages of the vessel, over a total distance of 50,000 miles.

A smaller meter was subsequently installed to measure small quantities of condensate from the winch condenser, so that the consumption of any auxiliary engine which could be separately exhausted to the winch condenser could be measured at any time. In the vessel in question this has proved a valuable means of locating losses, and has enabled important improvements to be made.

It is only such measurements that can give the required information. Measurements of the consumption of the auxiliaries in port can only be made under artificial conditions, and it is difficult to make these imitate the exact conditions when on service. At the same time it must be recognized the auxiliary consumption per shaft horsepower on one vessel under service conditions may not apply to another, and it is necessary to make a careful analysis of the figures.

Using as a guide the results of makers' trials, from which can be ascertained the consumption in pounds of steam per hour per air horsepower or water horsepower of each auxiliary engine, an attempt has been made in Table I to do this for the vessel described by Mr. Potts.

Table I.—Brake horsepower and steam consumption of auxiliary machinery for a geared turbine cargo steamer of 3230 shaft horsepower.

	Duty	Water H.P. or Air H.P.	B.H.P.	Estimated consumption lbs. per hour
Air pump	29,000 lbs./hr. condensate	....	5	350
Ejector		....	....	150
Circ-pump	27,000 galls./min., at 15 ft. head	12.4	18	882
Feed pump	2,600 lbs./hr., feed 200 lbs., b. p.	7.5	9	540
Fans	20,000 cu. ft./min. at 2" W.G.	6.4	11	580
Oil pump	6,500 galls./hr. at 50 lbs.	3.8	4.5	330
*Steering Engine				450
Electric light	2¼ K.W.	....	3.25	135
				3,415

As a confirmation of the general accuracy of this estimate, the total consumption for these auxiliaries ascertained by water meter as described above was 3450 pounds per hour.

The auxiliaries were supplied with steam at 100 pounds pressure; 100 degrees F. superheat and exhausted against a back pressure of 10 pounds gauge.

When the whole of the auxiliary exhaust steam can be condensed in the feed heater, it is possible to estimate the amount of the former by the rise in temperature of the feed water in passage through the heater, provided of course that the quantity of feed water is known. Where neither measuring tanks nor meters are fitted, this can always be ascertained approximately by

*The consumption of the steering engine is ascertained from measurements made with the small meter on the winch condenser condensate, referred to above.

means of an integrating counter on the feed pump strokes. Such an estimate of the auxiliary consumption however has to assume a value for the total heat per pound in the exhaust steam, which theoretically should correspond with the heat in the steam supplied to the auxiliary engines less the heat equivalent of the output of these engines, with due allowance for radiation and mechanical losses, but actually where measurements have been made is found to be considerably less, and of somewhat uncertain value. This method cannot therefore be considered to afford more than a rough estimate.

In twin screw vessels where tanks or water meter are provided for the separate measurements of the condensate on each side, a good determination can be made by exhausting all the auxiliary steam into one condenser for the first half of the trial, and into the other for the second half. This method is frequently adopted in naval vessels.

On the same basis as in Table I, a summary has been prepared of the estimated consumptions for the auxiliary plant of a 5000 shaft horsepower installation with high efficiency turbines.

Four different methods of driving the auxiliaries have been considered:

1. Entirely by steam.
2. By electric motors, with current supplied from turbo-generators.
3. By electric motors, with current supplied from Diesel-generators.
4. Mechanically from the main engines, with the provision of suitable "standby's."

This summary is given in Table II.

With a boiler pressure of 500 pounds per square inch, superheat to a temperature of 700 degrees F. and a condenser vacuum of 29 inches of mercury, the steam consumption of the main turbines is estimated at 7.2 pounds per shaft horsepower hour. The consumption and duties of the various auxiliaries have been based accordingly on a condensate of 36,000 pounds per hour.

Auxiliary exhaust steam at a pressure of 10 pounds above atmospheric is utilized for heating the feed water up to 215 degrees F. Where the auxiliary steam is insufficient for this purpose, the additional steam required is taken from the low pressure turbine steam belt. The feed water is further heated to a temperature of 310 degrees F. by steam withdrawn from the intermediate pressure turbine at a pressure of 110 pounds per square inch absolute.

In proposal (1) the steam consumption of the auxiliaries including the ejector steam, the make-up feed and the steam used for the evaporator is 8960 pounds per hour. Of this 1500 pounds is condensed in the evaporator coil, and 250 pounds in the oil fuel heater, leaving 7210 pounds available for feed heating. To heat the remainder of the feed water from 70 to 215 degrees F. requires 5800 pounds of steam, leaving 1410 pounds unutilized, which may either go to the winch condenser or be led to the low pressure turbine.

The total steam consumption for all purposes is, therefore, in this case 44,960 pounds per hour, or 8.99 pounds per shaft horsepower hour.

With a feed water temperature of 215 degrees F., 13.2 pounds of steam at 500 pounds per square inch and 700 degrees F. can be generated for each pound of oil consumed in the boiler furnace. This figure is based on a calorific value of 18,500 B.T.U. per pound and a boiler efficiency of 84 per cent, which can be attained with efficient pre-heating of the furnace air.

On this basis a steam consumption for all purposes of 8.99 pounds per shaft horsepower hour corresponds to an oil fuel consumption of 0.681 pound per shaft horsepower hour.

TABLE II.—SUMMARY OF THE ESTIMATED AUXILIARY CONSUMPTION FOR A 5000 S.H.P. HIGH EFFICIENCY INSTALLATION.

Auxiliary plant	Duty	Water H. P. or Air H. P.	B. H. P.	1. Steam driven Steam required lbs./hr.	2. Motor driven with Turbo generators. Steam required lbs./hr.	3. Motor driven with Diesel generators. 4. Driven mechanically from Main Engines. Steam required lbs./hr.
Air pump	36,000 lbs./hr. condensate, 29" vacuum 60° sea	....	7	450	....	....
Air ejector		....	....	200	200	200
Circulating pump (compound engines)	4,700 gallons per minute, 15 ft. head	21.5	31	1,200	....	....
Main boiler feed pump	44,960 lbs./hr. 500 lbs./sq. inch pressure, 700° F.	26.5	31	1,600	....	....
Forced draft fans	25,000 cub. ft./min. at 3" W.G.	12	20	860	....	....
Forced lubrication pump	10,000 gallons/hr. 50 lbs./sq. in. pressure	6	7	450	....	....
Oil fuel pump	3,400 lbs. of oil/hr. 200 lbs./sq. in. pressure	....	1	200	....	....
Steering engine	(36 B.H.P. motor)	....	8	750	....	....
Electric light engine (compound)		....	10	500	....	....
Bilge, sanitary and general service pumps		....	3.5	250	....	....
Make-up feed		....	....	1,000	1,000	900
Evaporator		....	....	1,500	1,500	1,350
			118.5	8,900	2,700	2,450
Steam to turbo-generator	110 K.W. (at 45 lbs. per K.W. hour)			....	4,950	....
Total auxiliary consumption				8,960	7,650	2,450
Condensed in evaporator coil				1,500	1,500	1,350
Condensed in oil fuel heater				250	250	250
Available for feed heating				7,210	5,900	850

By utilizing, however, steam tapped off from the turbines to raise the feed temperature to 310 degrees F., while slightly more steam is required, this is generated at a higher evaporative rate and the oil consumption per shaft horsepower is reduced to 0.667 pound.

The total brake horsepower required for the auxiliary machinery as enumerated in Table II is 118.5, and in proposals 2 and 3 this power is supplied by electric motors, absorbing with a reasonable allowance for motor and transmission losses, 110 kilowatts of electric power.

In proposal 2 the 110 kilowatts are supplied by turbo-generators taking steam at the same pressure and temperature as the main turbines, but exhausting to a back pressure of 10 pounds above atmosphere. At a consumption of 45 pounds per kilowatt they require 4950 pounds of steam per hour.

With the same supply of steam as before for ejectors, make-up feed and evaporator, viz., 2700 pounds the total auxiliary consumption in this arrangement is therefore 7650 pounds per hour, giving a total consumption for all purposes of 43,650 pounds per hour, or 8.73 pounds per shaft horsepower hour.

In this second proposal with 1500 pounds per hour condensed in the evaporator coil, and 250 pounds in the oil fuel heater, there remains 5900 pounds per hour of auxiliary exhaust steam which is sufficient to heat the feed water to 215 degrees F.

The steam consumption of 8.73 pounds per shaft horsepower hour for all purposes then corresponds to an oil fuel consumption of 0.662 pound per shaft horsepower hour, on the same basis of boiler efficiency, etc., which is further reduced to 0.648 pound per shaft horsepower hour by using tapped off steam to heat the feed water to 310 degrees F.

In proposal 3, the 110 kilowatts required for the motors is supplied by Diesel generators.

The additional steam for ejectors, make-up feed and evaporators is in this case 2450 pounds per hour with only 850 pounds per hour available for feed heating. Since, however, in this case 4950 pounds per hour is required to heat the feed to 215 degrees F., this 850 pounds must be supplemented by 4100 pounds of steam tapped off from the low pressure turbine belt, at the expense of a reduction of the power developed in that turbine by 285 shaft horsepower.

At a consumption of 0.75 pound of oil per kilowatt the Diesel generators consume 82.5 pounds of oil per hour, which at an evaporative rate of 13.2 pounds per pound of oil is equivalent to 1090 pounds steam per hour.

So that finally for this proposal 4715 shaft horsepower are obtained for a total equivalent consumption of 39,540 pounds of steam for all purposes, or at the rate of 8.39 pounds per shaft horsepower.

The corresponding oil consumption is 0.635 pound per shaft horsepower hour with feed temperature at 215 degrees F. and 0.62 pound with a feed temperature of 310 degrees F. using steam tapped off from the turbines.

In proposal 4 the 118.5 brake horsepower required for the auxiliary machinery is assumed supplied mechanically from the main turbines. With allowance for mechanical losses, it has been assumed that the output of the main turbines will thereby be reduced 132 shaft horsepower. As in proposal 3, it will also be necessary to bleed 4100 pounds of steam from the low pressure turbine for feed heating, involving a loss as before of 285 shaft horsepower. So that under this arrangement we have as the cost of supplying the auxiliaries, steam consumption increased by 2450 pounds per hour and output reduced by 417 shaft horsepower, or a total consumption of 38,450 pounds per hour for an output of 4583 shaft horsepower, or 8.39 pounds per shaft horsepower.

TABLE III.—EVALUATION OF OIL FUEL. CONSUMPTIONS OF 5000 S.H.P. INSTALLATION.

Conditions:

Initial pressure	500 lbs. per square inch (absolute)
Initial temperature	700 degrees F.
Condenser vacuum	29 inches mercury (Bar. 30 inches)
B.T.U. per lb. of oil	18,500
Boiler efficiency	84 per cent

	1. Steam driven Auxiliaries		2. Steam-electric Auxiliaries.	3. Diesel-electric Auxiliaries.	4. Auxiliaries driven from main engine.
	A.	B.			
Turbine Steam.....(pounds per hour)	36,000	36,000	36,000	36,000	36,000
Total steam for Auxiliaries....(pounds per hour)	8,960	8,960	7,650	2,450	2,450
Available for feed-heating.....(pounds per hour)	7,210	7,210	5,900	850	850
Utilized for feed-heating.....(pounds per hour)	5,800	5,800	5,700	4,950	4,950
Utilized in L. P. turbine.....(pounds per hour)	....	1,410	.....	.....	.....
Increase of S.H.P.....	....	98	.....	.....	.....
Tapped off from L. P. turbine..(pounds per hour)	....	....	.....	4,100	4,100
Decrease of S.H.P.....	....	....	.....	285	285
H. P. absorbed by Auxiliary.....	....	....	.....	.....	.....
Machinery in (4).....	....	....	.....	.....	132
S.H.P. for propeller shaft.....	5,000	5,098	5,000	4,715	4,583
Total Steam.....(pounds per hour)	44,960	44,960	43,650	38,450	38,450
Oil for Steam production.....(pounds per hour)	3,405	3,405	3,510	2,910	2,910
Oil for Diesel Generator.....(pounds per hour)	....	....	.....	82.5	.....
Total Oil.....(pounds per hour)	3,405	3,405	3,310	2,992.5	2,910
Lbs. oil per ∫ at 215 degrees F. feed temperature.	.681	.668	.662	.635	.635
S.H.P. hour ⟩ at 310 degrees F. feed temperature.	.667	.655	.646	.62	.62

This is the same as for proposal 3, and the oil fuel consumed per shaft horsepower is also the same as for proposal 3.

The calculations leading to these results are given in tabular form in Table III.

Summarized they are as follows:

For proposal 1, with steam driven auxiliaries, 0.667 pound oil per shaft horsepower hour;

For proposal 2, with motor driven auxiliaries and turbo generators, 0.649;

For proposal 3, with motor driven auxiliaries and Diesel generators, and for proposal 4, with auxiliary machinery mechanically driven from the main engine 0.62.

It will be seen from this comparison that with steam-electric auxiliaries the total fuel consumption may be made 2.85 per cent less than when they are steam driven, and with Diesel-electric or mechanically driven, 7.6 per cent less.

In proposal 4 it has, for the sake of simplicity, been supposed that all the auxiliaries are driven from the main engines. This may not be considered desirable in the case of all of them, even with the provision of adequate stand-by units. The result, however, as regards fuel consumption being the same as when these auxiliaries are motor driven from Diesel generators, one can make it apply equally to any combination of the two systems, that is to say, with as many of the auxiliaries as is considered practicable driven mechanically from the main turbines, and the rest motor driven from Diesel generators.

It has been assumed in determining the amounts of the steam consumptions that the steam supply to the auxiliaries is at a pressure of 200 pounds per square inch and saturated. If this steam is generated in a separate low pressure boiler it will absorb less heat per pound than the steam generated in the main boilers at a pressure of 500 pounds and 700 degrees F. The oil consumptions have, however, all been evaluated on the assumption that all the steam is generated in the main boilers.

To provide additional low pressure boilers for the auxiliary machinery, with duplication for the sake of overhauling and cleaning in port, would entail considerable addition to the plant. Some such provision is, however, necessary for the supply of deck machinery for cargo handling, and for the supply of saturated or only slightly superheated steam to the engine room auxiliaries before the main machinery is under way.

This difficulty can be satisfactorily overcome by the use of an attemperator as proposed by Messrs. Babcock and Wilcox. Only main boilers are fitted, and the steam required for auxiliaries is drawn from them and passed through the attemperator, in which it is intimately mixed with cold feed water, evaporating the latter, and its temperature thereby reduced. Thereafter it can be reduced by reducing valves to any pressure that is desired.

This simple method meets the practical requirements and is at the same time economical. For since some further steam is generated by the evaporation of feed water it will be seen that the steam required for the auxiliaries is obtained at a lower cost of heat per pound than in the main boilers. It will lead therefore to slightly lower figures for the total oil consumption than given in Table III.—"Marine Engineering," Aug., 1926.

DIPHENYL OXIDE BI-FLUID POWER PLANTS.

By H. H. Dow.

The object of the author in presenting this paper is to give particulars regarding a system that, it is believed, will convert a little larger percentage of the heat in coal into mechanical power than has been done by any system

heretofore. There is indeed a system which will convert the heat in oil through a Diesel engine and which, using the waste heat from the Diesel engine in steam boilers, will show a higher thermal efficiency than the one about to be described, which is based on the use of coal (which is very much cheaper per B.T.U. than oil), so that the two are not comparable from the dollars-and-cents standpoint.

ESSENTIAL REQUIREMENTS OF EFFICIENT POWER PLANTS.

It is well known that, according to thermodynamic principles, the maximum efficiency of conversion of heat into power that can be obtained between given high and low temperature limits is that of the Carnot cycle. The efficiency according to this cycle is equal to

$$\frac{T_1 - T_2}{T_1}$$

where the temperatures are measured on the absolute scale and where all the heat is put into the working fluid at the highest temperature T_1 and removed from the fluid at the lowest temperature T_2. For example, half of the heat energy in a fuel could be converted into power in a perfect engine if a fluid having no specific heat were boiled at 600 degrees F. and condensed at 70 degrees F. The Carnot-cycle efficiency would be

$$\frac{(600 + 460) - (70 + 460)}{600 + 460}$$

or 50 per cent.

Seventy degrees is taken as a starting point here, for it is about the lowest temperature to which a fluid can be expanded in a power plant without the use of an excessive amount of condensing water.

We are also limited at the upper end of the temperature scale, and, until the advent of the new chromium steel, it was impossible for a superheater pipe to withstand more than 725 degrees F. without deterioration. We know that at 750 degrees F. there is no creep of the metal. We know that at 900 degrees there is a creep. Whether there is a creep at 800 degrees or not we are not quite sure, but we do know that when we attempt to operate an ordinary superheater at 750 degrees, it will occasionally go enough above 800 degrees so that there will be a creep in the pipe, which means abnormal depreciation.

In Table 1, even where we get the high efficiencies of 80, 93, and 95 per cent in certain steps, the Carnot-cycle efficiency is only 45 per cent. All these other steps are so near 100 per cent that there is little opportunity of improving them. For this reason we may conclude that practically the only way of getting more horsepower from a heat unit is to go to a higher temperature at the upper end of the heat cycle, that is, increase the Carnot-cycle efficiency. This requires that all of the heat be put into the fluid at the upper temperature limit of the cycle, which has not always been fully realized by engineers in general. In a boiler plant provided with an economizer they will consider the B.T.U. that go into the economizer in the same sense as if they had gone into the original boiler; this despite the fact that the Carnot-cycle efficiency of the heat that goes into the economizer is not over 20 per cent, whereas the Carnot-cycle efficiency of the heat that goes into the boiler at 600 degrees F. is 50 per cent.

SOURCES OF LOSS DUE TO NOT PUTTING IN HEAT AT TOP OF CYCLE.

We are also limited in many other ways. For example, the modern steam plant, with its steam superheated to 750 degrees F., does not absorb the bulk of the heat from the coal at that temperature. If the steam pressure were in the neighborhood of 700 pounds per square inch, the water would boil at about 500 degrees F., and most of the heat from the coal would be utilized in boiling the water at this temperature. We should then have a Carnot-cycle efficiency corresponding to 500 degrees as the upper limit and 70 degrees as the lower limit, which is approximately 45 per cent. This would be increased only a moderate amount by superheating, because the amount of heat absorbed as superheat is relatively small.

This 45 per cent is decreased very materially in an ordinary plant because part of the heat in the coal has to raise the temperature of the condensate water from 70 degrees to the boiling point at 500 degrees. There is a way, however, to circumvent this loss, namely, by the so-called bleeder system, whereby steam that has already been utilized to a certain extent in a steam turbine and has lost some of its heat, is extracted therefrom to heat the boiler feed-water. Here only heat that has been put in at the top of the heat cycle is utilized to raise the temperature of the condensate nearly to the temperature at which the water is boiling. This method of bleeding, if carried out in an infinite number of steps, would make the Rankine cycle and the Carnot cycle exactly the same.

In practice, the method can be carried out only imperfectly; that is, the B.T.U. that are absorbed by the boiler have only about 90 per cent of the value they would have if they were absorbed at 500 degrees in the examples cited. This 90 per cent is the ratio of the Rankine efficiency to the efficiency theoretically possible under the working temperature if all the heat were put in at the top of the heat cycle, namely, at 500 degrees.

The other loss in power plants, namely, that incurred in heating the air from the outside temperature to that of the stack, can be largely eliminated by a heat interchanger, whereby the stack temperature may be reduced to within 100 degrees of that of the surrounding air, and the heat of the air fed into the furnace can be increased by the amount of heat that the stack gases lose.

PROPOSED REMEDIES.

Air Preheating. This method of increasing the efficiency of power plants, although well known and applied in the steel industry for at least two generations, has not been utilized to any extent in power-plant practice until very recently. In no case, so far as the author is aware, has it been used to heat the air under the grates to a temperature higher than 300 degrees F. There are a few examples where preheated air has been used for a good many years, mostly in British-built steamships where air preheaters in the stacks have recovered a certain amount of heat. But there are certain practical objections to utilizing air at a temperature of more than 300 degrees for combustion, and one of the objects of the present paper is to describe a practical way of overcoming one of these objections.

The author's company has developed a heat interchanger at its Midland plant that operates on the same principle as the heat regenerator in the Siemens open-hearth steel furnace, except that scrap iron is used instead of brick filling. There is a great deal of old iron pipe at the plant, and this has been cut to 4- and 5-inch lengths and then piled into the heat interchanger. This pipe scrap is first heated by the stack gases, and then, through an arrangement of valves, the flow is reversed and the heat in the iron is given up to the air that goes under the grate.

There is a vertical shaft through the center of the heat interchanger that operates the valves at the rate of one revolution every seven minuts. The interchanger takes air from the furnace and discharges it into the atmosphere at about 100 degrees above the temperature of the surrounding air. Then it takes in fresh air, which recovers a very large percentage of the total heat, and carries it down into the furnace. In fact, theoretically, it can put more degrees into the entering air than it took from the outgoing gases, because of the fact that the outgoing gases contain a larger percentage of steam—due to combustion of the hydrogen in the coal.

The main feature about such an apparatus is this: Instead of absorbing the lower-temperature heat in economizers, where the Carnot-cycle efficiency with which that heat is used is only 15 to 20 per cent, the heat of the stack gases is returned to the furnace to be absorbed by the boiler at the highest terperature of the cycle.

In an ordinary power plant, it is the author's experience—and that of many others—that 14 per cent CO_2 gives decidedly more trouble than 13 per cent. At 13 per cent CO_2 the heat is distributed through one-thirtieth more air than at 14 per cent CO_2, with the result that the temperature in the furnace is not as high. In the author's experience a lower power cost is obtained by running at 13 per cent CO_2 than at 14 per cent. There is a slight economy in coal consumption in going from 13 to 14 per cent CO_2, but it is more than counterbalanced by the increased repair bill.

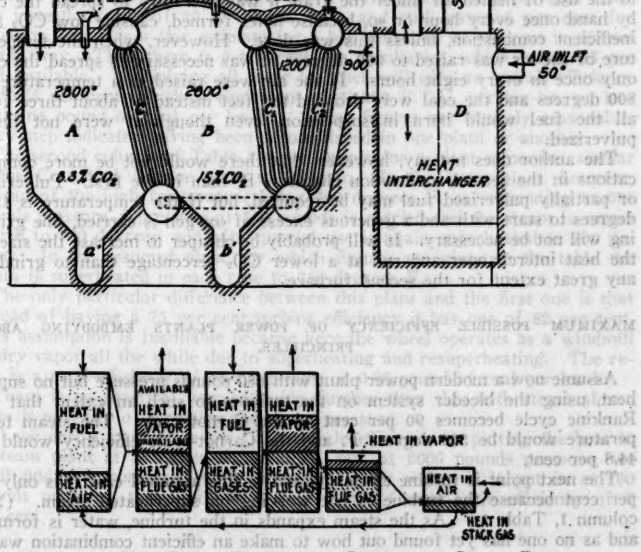


FIGURE 1.—TWO-STAGE COMBUSTION SYSTEM FOR STEAM BOILERS.

Feeding preheated air at 800 degrees F. and then going to 13 or 14 per cent CO_2 in a furnace would add the 800 degrees to the temperature now attained with disastrous results. For this reason the author considers it necessary, or at least advisable, to adopt the following system of combustion.

Two-Stage Combustion. In this system, referring to Figure 1, one-half the amount of coal to be burned is fed into furnace A at the top. Here it meets air from E at 800 degrees and it can well be imagined that the combustion will be rapid and good, as there will be a great excess of oxygen.

The gases then flow between the tubes C, where their temperature is reduced to 1200 degrees F. Next they meet the remainder of the coal coming into the second combustion chamber, B, and the temperature is again raised to 2800 degrees, assuming that there is no radiation to the tubes. Thence the gases pass on, as in other boilers, through the tube banks C_1 and C_2, discharging at 900 degrees into the heat interchanger D. A 900-degree stack-gas temperature would be quite fatal ordinarily—there would be a 20 per cent stack loss; but by using the heat interchanger and bringing back most of the heat, this loss can be brought down to 2 per cent.

A 2 per cent stack loss seems incredible, but it has been kept down to that in one installation at the author's plant for more than a year. The latest report on a second heat interchanger was 180 degrees F. stack temperature, the boiler operating at 200 per cent of rating. In this case a waste-heat boiler operated at atmospheric pressure to bring the temperature down as low as possible so that the temperature of the air going under the grate would not exceed 300 degrees F. because the stoker builders would not guarantee their stoker if that temperature was exceeded.

In the first interchanger installation a Newman stoker is used, the coal being blown in at a height of about three or four feet above the grate. Prior to the use of heated air under the grate it was necessary to spread the coal by hand once every hour or so, because holes formed, causing low CO_2 and inefficient combustion, unless this was done. However, when the temperature of the air was raised to 300 degrees, it was necessary to spread the coal only once in every eight hours. If the air were raised to a temperature of 800 degrees and the coal were dropped ten feet instead of about three feet, all the fuel would burn in suspension, even though it were not finely pulverized.

The author does not say, however, that there would not be more complications in the second combustion chamber, B, than in the first. Pulverized or partially pulverized fuel may be required, but if the temperature is 1200 degrees to start with and a generous excess of oxygen is carried, fine grinding will not be necessary. It will probably be cheaper to increase the size of the heat interchanger and run at a lower CO_2 percentage than to grind to any great extent for the second furnace.

MAXIMUM POSSIBLE EFFICIENCY OF POWER PLANTS EMBODYING ABOVE PRINCIPLES.

Assume now a modern power plant with 680 pounds pressure but no superheat, using the bleeder system on the turbines to such an extent that the Rankine cycle becomes 90 per cent of the Carnot cycle. The steam temperature would be 500 degrees F. and the Carnot-cycle efficiency would be 44.8 per cent.

The next point is turbine efficiency, and in this assumed case it is only 75 per cent because the turbine is not working on superheated steam. (See column 1, Table 1.) As the steam expands in the turbine, water is formed, and as no one has yet found out how to make an efficient combination water

TABLE 1.—COMPARATIVE EFFICIENCIES OF STEAM AND DIPHENYL OXIDE-WATER POWER PLANTS.

	Steam Plants		
	500 to 70 degrees F. (per cent) (1)	500 to 70 degrees F., superheat twice to 750 degrees F. (per cent) (2)	Diphenyl Oxide-Water, 750 to 70 degrees F. (per cent) (3)
Carnot-cycle efficiency	44.8	45.4	54.6
Ratio of Rankine to Carnot cycle	90	90	90
Steam-turbine efficiency	75	82	83
Generator efficiency	96	96	96
Boiler efficiency	93	93	93
Percentage of power available. (Total less percentage for auxiliaries)	95	95	94
Final thermal efficiency	25.65	28.4	34.2
B.T.U. per kw-hr	13,250	12,000	10,000
Coal per kw-hr., lb	0.95	0.86	0.72
Oil per kw-hr., lb	0.74	0.67	0.35

wheel and windmill, the efficiency is naturally poor. The generator efficiency is 96 per cent, which is common in big generators. The boiler efficiency is 93 per cent. This may seem too high, but it corresponds to a 2 per cent stack loss instead of one of 10 or 15 per cent, and still leaves 5 per cent for radiation, which is a fair estimate for a big boiler, so 93 per cent is not out of the way. Then there is 5 per cent for auxiliaries, which leaves 95 per cent available. Multiplying all these together will give an overall efficiency of 25.65 per cent, which is equivalent to 13,250 B.T.U. per kw-hr., or less than one pound of coal per kw-hr. These figures are entirely reasonable, every step indicated having been accomplished in one plant or another.

The second column of Table 1 gives figures for a plant somewhat similar to the Crawford Avenue plant in Chicago. In the latter plant, however, instead of the high temperature being 500 degrees F. corresponding to 681 pounds pressure, a pressure of 550 pounds is carried and the steam is superheated to 700 degrees, and after partial use is again superheated to 700 degrees. In the assumed plant for which figures are given in the table the steam is superheated in each case to 750 degrees F.

The only particular difference between this plant and the first one is that instead of having a 75 per cent turbine efficiency it has one of 82 per cent. This assumption is justifiable because here the wheel operates as a windmill on dry vapor all the while due to superheating and resuperheating. The result is a total efficiency of 28.4 per cent, or 0.86 pound of coal per kw-hr.

There is no practical way of going further than this with steam alone; there is no gain in increasing the steam pressure.

If we desire to increase the Carnot-cycle efficiency by putting the heat in a steam plant at a higher temperature, say, at 2000 pounds pressure, we shall find that compared with the steam boiler where it is put in at 700 pounds pressure, there will be a gain in the Carnot-cycle efficiency of 7 per cent.

Of that 7 per cent, 2 per cent will be lost because of the larger boiler-feed pump required. This will bring the gain down to 5 per cent. There will also be a loss because of the ratio of the Rankine cycle to Carnot cycle not being 90 per cent. That is, the same proportion of investment for bleeding and preheating will not give 90 per cent, but, say, 85 per cent, if working on 2000 instead of 700 pounds pressure. That will then wipe out the remaining 5 per cent and result in no gain at all.

It can be mathematically demonstrated that there is no other method available to materially increase the above-mentioned 28.4 per cent overall efficiency except by the use of a liquid that has a boiling point that is higher than that of water.

DIPHENYL OXIDE AND ITS PROPERTIES.

Mercury has properties that are superior in many respects to those of any other substance. It is capable of being raised to a higher temperature than any other known liquid. Any organic material has a limit beyond which it will decompose, whereas there is no temperature at which mercury will not be mercury. Accordingly, it has an unlimited field, if a material can be found that will hold and use it at extreme temperatures. This was clearly understood by the man who developed the bi-fluid system, Dr. Emmet. However, he failed to recognize the availability of the organic compound, which, so far as the author knows, has the highest boiling point with the greatest practicability, namely, diphenyl oxide. On the contrary, Dr. Emmet stated that it would decompose in a boiler. Now, the author's company happens to be the only manufacturer in the United States that makes diphenyl oxide on a large scale, and in its process of manufacture it employs a higher temperature than that at which Dr. Emmet thought it would decompose.

Accordingly, some of this diphenyl oxide was purified at various temperatures and it was found that it did not decompose under the proposed power-plant conditions, although it does if the temperature is made high enough, and the time long enough.

Diphenyl oxide is a white solid at room temperature, but a liquid if it contains a slight amount of impurity. It melts at 81 degrees F. when very pure. The fact that it is a solid is a slight objection, but any part of the boiler room where it would be used would have a temperature above 81 degrees. Its specific gravity is 1.083 and its boiling point at atmospheric pressure is 258 degrees C. or 496 degrees F., which is the temperature of saturated steam at 680 pounds pressure.

At 200 pounds pressure its boiling point is 800 degrees F. Its specific heat is approximately 0.4. Its critical pressure is 465 pounds at a temperature of 530 degrees C. or almost 1000 degrees F. Another physical property not often recorded in the tables is its price. In quantity lots it is worth 30 cents a pound. It can be produced in unlimited quantities, and the cost per unit volume is less than 2 per cent of the cost of the same volume of mercury. This is one principal reason for advocating diphenyl oxide in bi-fluid power systems.

One of the thermodynamic properties of diphenyl oxide is that in an ordinary turbine where it expands adiabatically, instead of condensing like water it superheats to a marked degree, even in a 100 per cent efficient turbine. For example, when the vapor expands from 135 pounds gauge down to atmospheric pressure where the boiling point is 500 degrees F., the temperature, instead of being 500 degrees, is about 625 degrees.

This extreme amount of superheating has an advantage and a disadvantage. It is difficult to transfer heat from a superheated vapor. Accordingly, when

this vapor is exhausted at 625 degrees it is necessary to use it regeneratively to heat the boiler feed in order to get 90 per cent of the Carnot cycle, otherwise the efficiency of this cycle would be only 75 per cent. We have 15 per cent as the goal to be gained by using regenerative heating, that is, taking the exhaust superheat from the turbine and using it for reheating the boiler feed, which corresponds to bleeding in an ordinary steam installation. A certain amount of this can be done, but whether or not it can be brought up to 90 per cent without prohibitive expense, has not yet been demonstrated.

Another advantage of diphenyl oxide is that the weight of its vapor is about 9.4 times that of steam vapor. This means that a diphenyl-oxide turbine will run at a lower speed with higher torque for the same horsepower than a steam turbine, which is a great advantage. It may be that on that account we shall get a little higher turbine efficiency, and we shall certainly get some higher turbine efficiency by reason of the fact that there is never any condensate in the vapor.

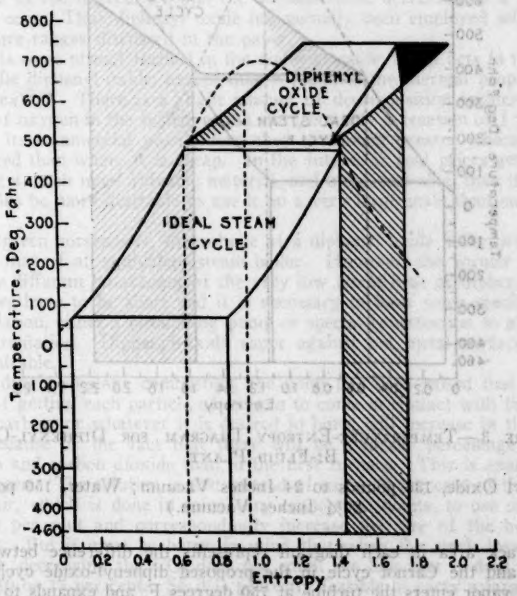


FIGURE 2.—TEMPERATURE-ENTROPY DIAGRAM FOR DIPHENYL-OXIDE BI-FLUID PLANT.

(Diphenyl Oxide, 135 Pounds to Atmosphere; Water 600 Pounds to 29¼ Inches Vacuum.)

One difference between diphenyl oxide and mercury is that the latter does not wet steel, and on this account the boiler tube may get much hotter than the mercury; whereas diphenyl oxide does wet the tube, so that it is easier to transfer the heat from the boiler to the liquid.

For these various reasons the author feels very confident in proposing the diphenyl-oxide bi-fluid plant working between 750 and 70 degrees F. The diphenyl-oxide vapor would be generated in a boiler at 135 pounds gauge pressure and expanded to a vacuum depending on the steam pressure with which it is desired to work. The temperature-entropy diagrams show two such combinations, one (Figure 2) in which the diphenyl oxide is condensed at atmospheric pressure, making steam of about 600 pounds pressure, and the other (Figure 3) exhausting at about 24 inches of vacuum, making steam of 150 pounds pressure.

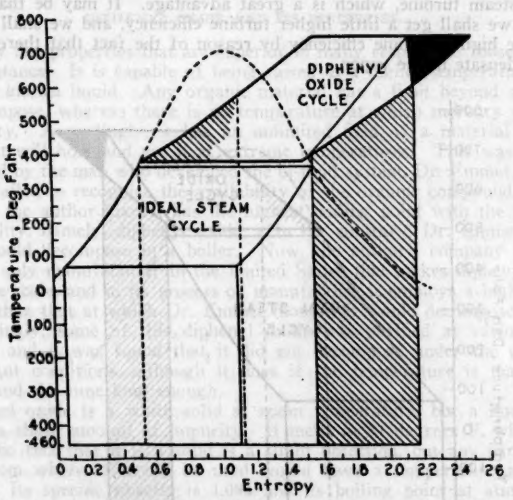


FIGURE 3.—TEMPERATURE-ENTROPY DIAGRAM FOR DIPHENYL-OXIDE BI-FLUID PLANT.

(Diphenyl Oxide, 135 pounds to 24 Inches Vacuum; Water, 150 pounds to 29¼ Inches Vacuum.)

The black area in each diagram represents the difference between the Rankine and the Carnot cycle in the proposed diphenyl-oxide cycle. The saturated vapor enters the turbine at 750 degrees F. and expands to the exhaust pressure with a temperature considerably higher than the condensing temperature. The superheat in the vapor, represented by the shaded area, is then removed in a counterflow boiler-feed preheater. The heat of vaporization at the condensing temperature is given up to the water in the condenser boiler and the liquid diphenyl oxide is returned to the boiler through the preheater. The steam should be superheated by direct heat or by diphenyl-oxide vapor in order to gain the increase in the efficiency of the turbine.

* * * * * * * *

Following the presentation of his paper Dr. Dow replied to questions propounded from the floor. The additional information thus brought out is summarized below.

Superheating of the exhaust vapor is a thermal property of diphenyl oxide and many other organic substances. Whenever the heat put into the liquid plus the heat of vaporization is in excess of the heat that can be turned into mechanical energy by a turbine plus the total heat of the saturated vapor at the condensing pressure, the difference goes into superheat. This is the case with all liquids that had a high ratio of specific heat to latent heat. The heat that goes in as specific heat into the liquid and into the latent heat of evaporation, comes back partly as power and partly as superheat in the vapor.

As to the instability of diphenyl oxide when heated, some of it was boiled in an iron boiler for a month at 830 degrees night and day. At this temperature there was some decomposition after a month—in the neighborhood of 1 or 2 per cent. It was then boiled at 135 pounds pressure, which corresponded to 750 degrees F., and the decomposition decreased to a fraction of 1 per cent. Thus diphenyl oxide has actually been employed within the temperature ranges discussed in the paper.

There is some phenol formed in the diphenyl oxide, but it acts in the same way as the diphenyl oxide, except that it modifies the thermal properties to a slight extent. There is a slight amount of decomposition incident to the amount of oxygen in the system at the start, but only a fraction of 1 per cent.

As to its commercial possibilities, they are much greater where coal is high priced than where it is cheap. In the future, as coal prices get higher, it will be a much more valuable material, and more desirable, than it is now. It will also be more desirable to use it on a very large scale than on a small one.

For a given horsepower, the volume of a diphenyl-oxide boiler will be the same as that of an equivalent steam boiler. However, the former boiler is distinctly different on account of the very low latent heat of diphenyl oxide. The tubes have to be short and it is necessary to have some special means of circulation, either a circulating pump or special construction to give maximum circulation. Diphenyl-oxide vapor against hot metal surfaces would be undesirable.

Regarding two-stage combustion, the point has been raised that the difficulty of getting each particle of oxygen to come in contact with the hydrogen or carbon, or whatever it is desired to burn, will increase in the second stage because of the fact that there will be a larger percentage of inert nitrogen and carbon dioxide than in the first furnace. This is exactly what will happen. The solution is that instead of using 72 per cent of the oxygen in the air, which is done in some very economical plants, to use only 68 or even 60 per cent and correspondingly increase the size of the heat interchanger. But it must be borne in mind that when the stack loss is but 2 per cent, decreasing the CO_2 does not entail the loss that it does when the loss is 10 or 15 per cent. So it is perfectly feasible to carry a larger excess of oxygen in the second stage than it would be provided there were no heat interchanger at the end.

The stack gases may be handled with an induced-draft fan, the use of which has increased the cost of operation about 0.4 per cent in the company's Midland plant. To take care of this, it would be noticed in Table 1 that 6 per cent has been allowed for auxiliaries instead of 5 per cent.

Diphenyl oxide has not as yet been tried in a turbine. As to heating up the turbine before starting, any turbine should be heated up, especially one operating at high temperatures. Of course, it depends on the amount of clearance, and if this is small then the turbine will have to be warmed slowly. External heating might be an advantage.

In addition to the experimental boilers used in the laboratory, a 1-horse-power diphenyl-oxide boiler has been tried out with good success. Diphenyl oxide has not been found to have any poisonous effects, its present use being in the perfume industry on account of its geranium odor.—"Mechanical Engineering," August, 1926.

STEAM PIPES FOR EXTRA HIGH PRESSURE AND TEMPERATURE.

By J. A. Aiton, C. B. E.

Some of the difficulties to be met with in laying out steam piping on land are either much smaller or are not met with in marine work at all, but on the other hand, due to the confined space on board ship, others are much exaggerated. For instance, on board ship the run of the piping cannot be varied to any great extent, although here also simplicity should ever be aimed at, but for the same reason the method of allowing for the movement of pipes due to expansion by means of wide bends in the run of the pipes cannot be adopted. For this reason, sliding expansion joints or ball joints have been largely used and have been fairly satisfactory with lower pressure saturated steam, but with superheated steam they are not satisfactory and have practically disappeared from land practice. We are thus thrown back on bends, but these must be of short radius and consequently so very stiff that excessive thrust comes upon the fixed points if any serious movement has to be allowed for, and great stresses are set up in the joints; moreover, bends made to short radius show quite appreciable thinning of the wall on the outside of the bend. Undoubtedly, the most serious difficulty which will be found in fitting pipes on board ship to carry high-pressure steam is this one of dealing with the movements due to expansion, and if the two methods of meeting the difficulty, mentioned above, were the only ones the position would appear to be very difficult indeed, but fortunately another method has been devised whereby bends are made which are much more flexible than those made from plain tube.

CORRUGATED PIPES.

This method is to form a series of corrugations on the tube—these in themselves strengthen the tube, especially as the process of manufacture thickens the wall of the tube. The corrugated pipe is then formed into a bend in the usual way, and there can be no thinning of the wall in bending, as all that happens is that the corrugations are somewhat opened out on the outside of the bend and are closed up on the inside. These corrugated pipes can be bent to a much smaller radius than plain pipe, and in extreme cases they can be bent to the same radius as an elbow. In all cases, to stand corrugation, the tubes must be of a good soft quality of steel and must be well made and free from variations of thickness. They are usually solid drawn tubes, hot finish, 24 tons to 28 tons tensile, 25 per cent elongation in 8 inches.

Bends made from corrugated tube are five times as flexible as similar bends made from plain tube, that is to say, if the same thrust is applied to both ends the corrugated one will move five times as far as the plain one, or if the same movement is required only one-fifth of the thrust required to move the plain bend need be applied to the corrugated one; or again, if the same amount of movement is required and the same amount of thrust can

be applied to both bends then the corrugated bend can be made to a much smaller radius, and it is this last quality which will be found so useful by the marine engineer. Comparison with the lyre type of expansion bend is interesting. Such a bend made from plain tube is 138 inches high from the center of the main to the center of the pipe at the crown of the bend, and allows for a movement of 2½ inches out or in. A bend made from corrugated pipe of the same bore can be made 55 inches high and gives 2¼ inches out or in, and a similar bend 72 inches high gives 3⅛ inches out or in, or 50 per cent more than the much larger plain bend.

There are other classes of expansion which can be at least minimized by care in design. Water in pipes is the cause of the most serious stresses. It is not possible entirely to prevent it from getting into the pipes, but the amount which passes along with the steam does not do much harm. Water, however, which is allowed to collect and lodge in the pipes may do considerable damage. When steam is turned into horizontal pipes in which water has been allowed to collect, unequal expansion is set up in the pipes, hogging takes place, and stresses arise, the ultimate effect of which cannot be calculated.

Injudicious anchoring causes expansion stresses by endeavoring to force the pipe into a position where it would not naturally go. The ideal plan is to have no anchoring at all, as it is frequently only necessary to guide the pipe in the direction in which expansion should take place; but this ideal cannot always be obtained, in which case anchors must be fitted, but considerable experience is required to place them in the right position.

Referring to questions of the design and manufacture of the pipes themselves, undoubtedly the weakest part of the pipe line is the joint, and as every joint is a potential cause of trouble, the obvious aim should be to reduce the number of joints to a bare minimum by making each pipe as long as possible. As castings, even of steel, are no longer used in first-class pipework, and as modern methods allow branches to be fixed to any part of the pipe, it is possible to use straight pipes up to 35 feet or even more, and 30 feet is quite a common length. The entire elimination of joints by butt-welding pipes end-on in position is very attractive, and there is an instance at work in this country. This main is 6-inch bore, one straight length is 260 feet long, steam pressure is 350 pounds, and the temperature is 750 degrees F.; there are two anchors 200 feet apart, and the expansion is taken up by one corrugated lyre-type expansion bend 100 inches high. There is only one pair of flanges on the job and these are at the crown of the expansion bend. Care is taken to keep the pipe true to alignment by means of roller guides, which prevent cross stresses coming upon the butt-welded joints, and where, in spite of this precaution, there is any possibility of cross stresses coming upon the joint it is reinforced by having collars welded over the butt weld. It is early yet to say that this method of joining pipes will supersede flanges, but in special circumstances it will be found very useful.

PIPE FLANGES.

In the use of flanges for piping there are two main questions to consider, namely, how to attach the flange to the pipe, and how to make the joint between the flanges. Regarding the first, there are nowadays only two methods of fastening flanges to pipes, namely, by riveting and by welding, except in the case of small pipes, which nearly always have flanges fixed by screwing. It seems on the face of it to be wrong that holes should be made in a pipe to be afterwards closed by rivets. A satisfactory job, however, can be made in this way, but sudden variations in temperature may start a leak even

after the pipe has been in use for a long time, and if a rivet does leak under high-pressure high-temperature steam it is almost impossible to make it tight. Frequently the presence of rivet heads inside the pipe, coming very close together, seriously reduces the area of passage. With high-velocity steam this either means a continuous loss due to a greater pressure drop, or a larger-sized pipe to pass the same weight of steam.

Flanges properly welded to pipes are subject to none of the disadvantages of riveting, but unfortunately there are many methods of welding flanges, and they are not all satisfactory. The only satisfactory method of fastening flanges to pipes is by welding them on by means of the carbon-arc process. Unless the flanges are exceptionally thick, a continuous weld should be formed right through the flange from back to front, with a good fillet at the back.

Having fastened the flange to the pipe, the next important question is how to make the joint. The flanges must be heavy to minimize the probability of springing, the joint must be made on a raised face inside the bolt holes, and a ground face is very desirable, though not essential. By care, experience, and the use of good materials, joints made by the same methods as have been used for many years are giving satisfactory service today, that is to say, with an ordinary corrugated metal ring and a wash of thin cement. Many such joints have been in use for years, and have never been touched since they were made; but it must be added that in the original lay-out of the piping, expansion stresses must be reduced to a minimum and every care taken to prevent water collecting in the pipes. Expansion stresses and water are the two great enemies of pipe joints, and water is by far the worse.

No good joint can be made with the wrong type of bolt, and bolts made from steel of from 30 to 35 tons tensile strength have given most satisfaction. Bolts made from high tensile steel have shown a liability to brittleness after being subjected to variations of temperature.

Where branches have to be taken off pipes this should be done by means of welded-on stools, and experience has shown that the oxy-acetylene process is the most suitable for this work. At one time branches were welded on by the carbon-arc process, but, though to all appearance the weld was quite normal, it sometimes broke down under test, and as there was a possibility that one such branch might pass the test and yet break down in working, the risk was considered too great, so this process was given up. Oxy-acetylene welding is more expensive, but it has given no sign of failure in working. In branch welding there are only two thin pieces of metal to be dealt with. Therefore the period during which the full temperature of the carbon arc can be kept on the metal with safety is very short. This is possibly the root of the trouble, and if so, it is inherent in the process. The branch piece is not taken through the pipe, because it is necessary to get access to the joint from the inside both with the blow pipe and with the hammer, and as this can often only be done through the branch the joint must be inside the branch. Cast steel tees must be ruled out of this class of pipe-work owing to the great increase in the number of joints, and riveted stools, owing to the fact that they must usually be riveted by hand, are not often satisfactory, and as they can only be fixed near the end of the pipe, they cause a lot of otherwise unnecessary joints in the main. It is advisable to have as few joints as possible. With simple design and large units it should be possible to do away with branches, and to bring the pipes into vessels which serve as collectors and distributors as well as drainage points. Collector vessels should be placed with the axis vertical, for in this position it is easier to arrange the branches and the drainage, and vessels so placed are not subject to hogging stresses which are prevalent in vessels placed horizontally when water accumulates in them.

I may sum up by saying that no drastic alterations are necessary either in materials or in the method of manufacture to meet modern conditions in piping, but great care and experience are necessary in all departments of the work, especially in design.—"Shipbuilding and Shipping Record," Sept. 16, 1926.

A FRENCH HIGH-SPEED DIESEL.

By R. H. Andrews.

There is a movement among some French experts favoring the use of solid injection for all types of internal-combustion engines, from the high-powered Diesel down to the gasoline engine for automobile drive, regardless of the fuel employed. The advantages claimed include the possibility of using higher compression pressures, with the resulting improvement in efficiency. This is possible because the mixture of the injected fuel with the air is imperfect and, therefore, unfavorable to the production of the detonating wave, even with volatile fuels. In addition there is the possibility of using a much wider range of fuels. It is claimed by advocates of fuel injection that, in order to secure a satisfactory mixture with a carburetor, it is essential to use a fuel of approximately homogeneous physical constitution; that is, all the elements should vaporize over a limited range of temperatures, whereas injected fuels may include elements differing widely in boiling points. Whether these advantages will be sufficient to displace the carburetor remains to be seen. It does not seem likely, however, that injection will ever be used with gasoline alone.

LIGHTEST OIL ENGINE BUILT.

The Peugeot-Tartrais oil engine is interesting primarily as the lightest oil engine built, weighing only 11 pounds per horsepower, and it is first to undergo a successful road test in automobile drive. In October, 1922, a five-passenger touring car driven by this engine completed a 700-mile run in two days. The fuel consumed was gas oil, and the mileage per gallon of oil about 14¼ miles.

This engine is also particularly instructive, not only because of a number of novel features in its design, but because of the curious story of its development. It was originally planned as a semi-Diesel, or hot-bulb, engine built for the closest possible approach to constant-volume combustion, and was subsequently modified so as to bring it within the Diesel class.

The original idea of the inventor, Tartrais, was to develop a surface-ignition in which the mixture of air and oil vapor could be made sufficiently homogeneous before ignition to give a high rate of flame propagation. For this purpose the combustion chamber was made of peculiar shape and was connected to the cylinder proper by a bottle-neck opening. The piston face was provided with a projection, or nose, which, on dead center, extended about halfway through the bottle neck. The nose was so shaped that during the latter part of the compression stroke the air was forced at high velocity between the piston nose and the walls of the bottle neck and also through oblique holes drilled through the nose. This and the shape of the combustion-chamber walls produced a high state of turbulence during the injection of the fuel. The latter was injected mechanically in a horizontal plane at right angles to the axis of the bottle neck, so as to be swept up by the stream of inflowing air and rapidly mixed with it.

The compression pressure was about 280 pounds per square inch and the maximum pressure during combustion about 450 pounds. An electrically heated plug comprising a platinum resistance wire, was provided for starting. The model built had two cylinders and operated on the two-stroke cycle. The scavenging air was supplied at about 5 pounds gauge by a separate scavenging pump.

FUEL CONSUMPTION LOW.

The fuel consumption was reported as 0.4 pound of gas oil per brake horsepower hour, a surprisingly low figure hardly ever reached in Diesels.

Though technically satisfactory in most respects, the design proved so complicated as to make the production cost of the engine prohibitive. The

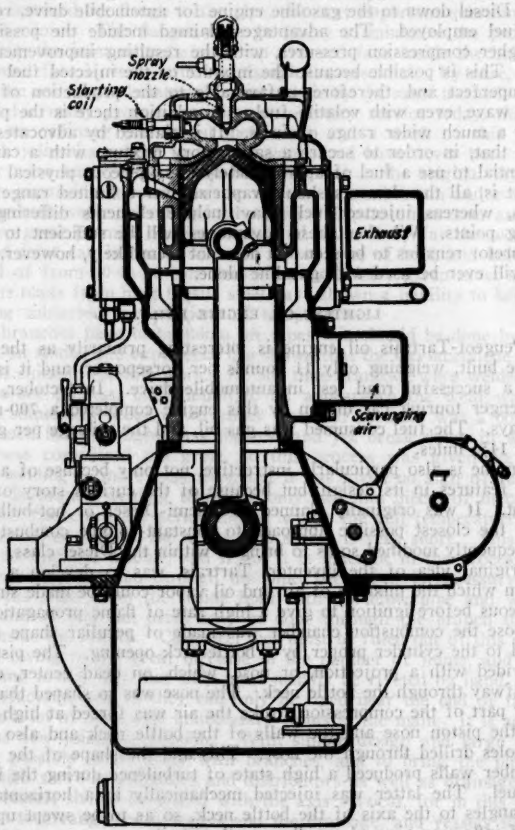


FIGURE 1.—CROSS-SECTION OF PEUGEOT-TARTRAIS ENGINE.

engine was then carefully gone over and it was first decided to water-cool the walls of the combustion chamber in order to increase their strength. This made it necessary to raise the compression volume ratio to 12 in order to bring the mixture to the temperature of auto-ignition. The resulting increase in the compression pressure to 475 pounds made a high degree of turbulence in the mixture undesirable, in order to avoid excessive combustion pressures. It was therefore possible to discard the nose on the piston face, resulting in a material reduction in the production cost of the latter.

These changes, together with the simplification of the treatment of the lubrication problem, constituted the principal modifications in the original design. The fuel consumption of the new engine is about 0.45 pound per brake horsepower-hour at full load. The smoothness and regularity of operation have not been impaired, and the reliability of the engine has probably been increased. The compression pressure being somewhat too low for cold starting, the starting plug was not discarded.

A cross-section of the new Peugeot-Tartrais engine, as it is to be placed on the market, is shown in Figure 1.

The only type to be built at first is a two-cylinder, two-stroke-cycle motor truck engine, 50 to 55 horsepower maximum at 1400 R.P.M. The cylinder dimensions of this engine are 4.73-inch bore and 5.906-inch stroke. The scavenging air is supplied by a separate piston-type air pump which is located between the engine and fan. The pump also produces a slight degree of supercharging, through the medium of a revolving valve placed at the outlet from the exhaust manifold, which is timed to close the exhaust when the cylinder is free of burnt gases.

When the Tartrais engine is running at rated speed, 1400 R.P.M., and under maximum load, each of the two pump cylinders must deliver 23 times per second a volume of about 0.004 cubic inches of oil, the maximum length of the delivery stroke being less than 0.08 inch. The volume is so small that the greatest care must be exercised to reduce to a minimum the compressibility of the oil, by avoiding the leakage of air into the fuel circuit, and particularly the formation of bubbles of vapor, under the extremely rapid fluctuations of pressure to which the oil is subjected.

The suction stroke of the fuel pump is altered to meet the load by a stop, which has a long-pitch screw thread and may be revolved by a worm and gear operated by the fuel-regulating rod. The time of injection may be adjusted by means of a sliding sleeve coupling connecting the camshaft to its driving gear through the medium of a helical groove. The Peugeot company is planning a series of engines for stationary and marine service, of 50 horsepower per cylinder in multiple-cylinder units.—"Power," Aug. 3, 1926.

NAVAL AVIATION.

To say that aviation is destined to play an important part in future warfare at sea is to utter a platitude. Aircraft have already become an integral component of every well-balanced fleet, and their unique value has been demonstrated in actual war operations. In the development of this new arm the British Navy may claim to have led the way. While there is some doubt as to whether an aeroplane was first flown from the deck of a British or an American man-of-war, it is not disputed that we were the first in the field with the aircraft carrier, which has made possible the remarkable progress in naval aviation. At the beginning of the war several cross-Channel steamers were converted into aeroplane carriers, one of which did

useful service at Jutland. Then the Cunard liner *Campania* was fitted with a flight deck, from which machines could take off, though they were unable to make landings on it. The next stage was the conversion of the light battle-cruiser *Furious*, which became the prototype of the modern carrier. She proved a great success, and it was from her deck that the most successful naval air attack of the whole war was launched, when a flight of machines bombed the German airship sheds at Tondern, destroying two Zeppelins. Meanwhile, the Admiralty had purchased the Chilean battleship *Cochrane* and the Italian liner *Conte Rosso*, lying unfinished in British shipyards, and caused them to be rebuilt as aircraft carriers, under the respective names of *Eagle* and *Argus*. Finally, in January, 1918, the *Hermes* was laid down, she being the first ship in the world to be designed *ab ovo* for the transport of aircraft. Since the war the two battle-cruisers *Glorious* and *Courageous* have been taken in hand for conversion. The result is that we now possess a fleet of carriers of which there is no counterpart in any foreign navy. Excluding two that are obsolete, it consists of six vessels, ranging from 10,550 to 22,600 tons. The *Argus*—the slowest unit—has a speed of just over 20 knots; the *Eagle* is good for 24 knots; the *Hermes* for 25; and the *Furious, Glorious* and *Courageous* for 31 knots. The combined aircraft capacity of these six ships is not definitely known, but it cannot be less than 160 machines.

Turning to other navies, the United States is first with two very large carriers, *Lexington* and *Saratoga*, originally begun as battle-cruisers. The displacement is 33,000 tons, the designed speed 33.34 knots, and the stowage capacity seventy-two aeroplanes per ship. Unlike our carriers, which mount nothing heavier than 6-inch guns, the American type is very powerfully armed, having eight 8-inch 55-caliber guns in twin turrets, twelve 5-inch quick-firers on high-angle mountings, and four torpedo tubes. It is, in effect, a composite type, half battle-cruiser and half aircraft carrier, its combatant character being further emphasized by substantial belt armor and bulge protection. Japan is pursuing much the same policy. Her largest carrier, the former battle-cruiser *Akagi*, of 27,000 tons and 28.5 knots, mounts no fewer than ten 8-inch and sixteen 4.7-inch guns, and it is understood that the *Kaga*, an ex-battleship, of 26,900 tons and 23 knots, will have a similar armament. It will be seen that H.M.S. *Eagle*, our largest ship, is smaller by 10,400 tons than the American type, and is considerably lighter than the Japanese vessels. In British naval circles, however, the utility of such monster ships is questioned. Logically considered, the carrier is not a fighting ship but an auxiliary; her prime function is to furnish a floating base for aircraft working with the fleet. Being peculiarly vulnerable to gunfire and other methods of attack, she should keep as far away from the enemy as is consistent with the performance of her special duties. Otherwise, the necessity of affording her protection will impose a serious burden on the fighting ships proper. Moreover, the sinking or disablement of a carrier such as the *Lexington* or the *Akagi* would be a disaster of the first magnitude, since it would deprive the fleet to which she belonged of half its air force at a single blow. Hence the preference in this country for carriers of moderate dimensions, such as the *Hermes*. The reconstruction of such comparatively large vessels as the *Glorious* and *Courageous* is probably dictated by motives of economy. Had funds been available, new ships of smaller tonnage would doubtless have been laid down. But, although American policy with regard to carrier dimensions may be open to criticism, there are other features of aviation development in the United States Navy which seem to us worthy of emulation. We refer particularly to the principle of supplying aeroplanes to every type of vessel capable of carrying

them. According to recent information, every American battleship in the fleet now carries one scout and two combat machines, and each cruiser two scouts. They are discharged from the ship by means of catapults. During the visit of the United States light cruiser *Memphis* to Gravesend this month we had an opportunity of inspecting her aircraft equipment. The two seaplanes were stowed amidships, each on its own catapult. Certainly they occupied a good deal of deck space, but their presence did not seem to interfere with the effective working of the ship or her armament. We were informed that flying exercises were part of the ordinary routine of training, and that no difficulty was experienced in launching the machines in normal weather. They are propelled from the catapult by an air blast at a velocity of 60 miles an hour. On returning from the air they alight on the sea as close to the ship as possible, whence they are hoisted in by the derrick. By constant repetition the operation of launching the machines and recovering them from the sea has been brought to a high standard of efficiency. Seaplanes of a diminutive type have been designed for the use of destroyers, and twenty of these boats now carry one machine apiece. Even submarines have been experimentally fitted with tiny seaplanes. In all, something like a hundred aeroplanes are now distributed among the ships of the United States Navy, in addition to those borne in the carriers. The battleships use their aircraft chiefly for "spotting" purposes, and are thus able to acquire practical experience in long-range indirect gunnery. Cruisers detailed for patrol or reconnaissance must find their aircraft invaluable. Had our cruisers been equipped with aeroplanes on the outbreak of war it is virtually certain that all the German raiders then at large would speedily have been laid by the heels. The career of the *Emden*, *Karlsruhe*, and others would have been numbered by weeks or even days, instead of by months. At that period, however, reliable aircraft for use on board ship had not been evolved, and so our cruisers had to dispense with what would assuredly have proved an adjunct of priceless value. But by now the most serious technical difficulties have been overcome, and it is to be hoped that the Admiralty will not be behindhand in providing aircraft for such of our warships as could employ them with profit, that is, battleships and cruisers.

The American Navy enjoys one marked advantage: it has absolute control of its air branch, which is thus organized strictly in accordance with naval needs, no other department having a voice in the matter. In time, as we hope, this rational system will be introduced here, and the Navy's flying service divorced from Air Ministry control, which acts as a drag on the wheels of progress. Convincing evidence of the importance attached to aviation by the American naval authorities is afforded by recent legislative action. A bill which passed through Congress last month, and has since become law, authorizes the expenditure of £17,000,000 on the naval air service during the next five years. It provides for the construction of 1947 new aeroplanes and the maintenance at all times of an effective force of not less than 1000 machines. Lighter-than-air craft are to be represented by two rigid dirigibles, each to be three times as large as the ill-fated *Shenandoah*. Not the least important section of the bill is a clause which creates the post of Second Assistant Secretary of the Navy, whose sole duty will be the administration and supervision of the air arm. It is further enacted that every officer appointed to command an aviation school, a flying station, or an air unit shall be a qualified aviator. Thanks to this far-reaching measure, the future of the American Navy's air service would seem to be assured. On the other hand, though the present strength of the British Navy in aircraft is difficult to estimate in the absence of reliable figures, we believe we

are correct in describing it as inadequate even for the needs of the moment. There should, we submit, be enough machines on hand to fill up all available aircraft carriers to capacity and to provide at least two for every battleship and at least one for every cruiser on the active list, with a liberal margin in reserve. No doubt the Admiralty is doing its best to attain this standard, but there are obvious difficulties in the way, not the least of them being the well-meant efforts of the Air Ministry to balance the conflicting claims of the Navy, the Army and the Royal Air Force itself. We shall not presume to decide between the claims of these respective services, but we are definitely of opinion that the formation of an adequate and autonomous Naval Flying Corps is a national requirement of capital importance.—"The Engineer," Aug. 6, 1926.

THE MAUBLANC-LALLIE MARINE PROPELLER.

It has been generally accepted among propeller engineers that the parts of the blade which give a maximum of efficiency consist of surfaces inclined at 45 degrees to the axis of the propeller. What the present inventors did was to make an effort to develop to the utmost the employment of surfaces inclined at 45 degrees and thereabouts. It is stated that in the ordinary propeller, because of the elliptical contour of the blade, the region within the angle of 45 degrees is comparatively little developed.

Because of the effort to develop surfaces having an angle of inclination of the order of 45 degrees, the new propeller had to be given an unusual shape. In rotating it describes a truncated cone. It attacks the liquid with the thin edges of the blades, which latter gradually increase in width and thickness. The surfaces which are displaced near to the axis and ahead of the hub produce along the latter a very rapid flow of liquid, which inhibits cavitation. It has been observed that the wake with this propeller is very narrow, which proves that the propulsive effort is exerted along the axis with a minimum of sidewise projection of the liquid.

The new propeller has been installed in and tested on several vessels previously equipped with conventional propellers, and it is said that the substitution has given most gratifying results. On the steamer *Saint-Brevin* the old propeller had four blades, 1.66 millimeters (65.3 inches) in diameter and was replaced by a Maublanc-Lallié three-blade propeller 1.54 millimeters (60.6 inches) in diameter. The 260-horsepower triple-expansion engine ran the old propeller at 250 revolutions, while the new propeller at 212 revolutions gives the vessel the same speed as before. The reduction of revolutions means of course an economy in fuel oil and longer life of the engine.

In official tests of the gunboat *Tapageuse* (1300 horsepower at 300 R.P.M.) made with a 1.94 millimeter (76.3 inches) three-blade propeller, the results showed a gain in speed over what was formerly obtained with a conventional propeller of the same diameter.

The efficiency of the propeller in reversing was shown to be particularly good. Running at full speed ahead the vessel was stopped within one length.

Moreover, it was found that with this propeller vibration is considerably reduced. This was discovered in tests with the 500-ton tugboat *Centaure* (634 horsepower). With the conventional type of propeller this vessel shook in a distressing manner. When, however, the new propeller was installed, the vibrations were greatly reduced.

The proportions of the various parts of the propeller, shown in Figure 2, cannot be determined in a general manner, and each unit has to be designed with due regard to its operating conditions. (W. Lallié in "La Nature," No. 2721, May 29, 1926, pp. 351-352, 3 figs., *d.*)—"Mechanical Engineering," August, 1926.

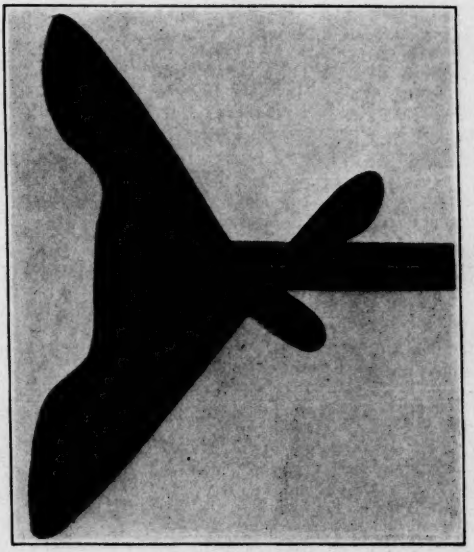
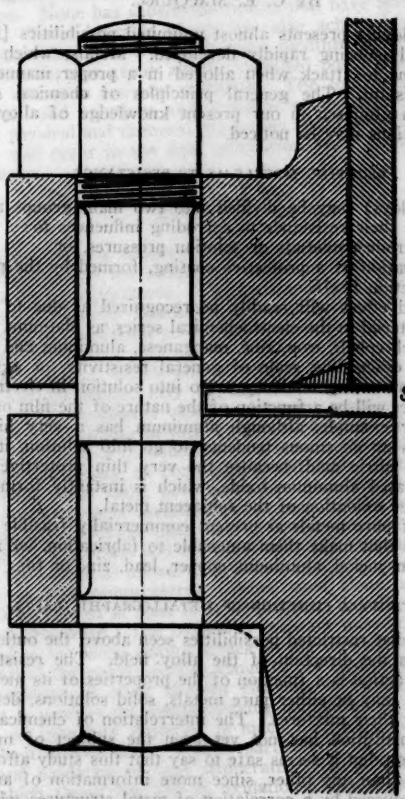


FIGURE 2.—TWO-BLADE MAUBLANC-LALLIÉ MARINE PROPELLER.

HIGH-PRESSURE PIPE JOINTS AT LANGERBRUGGE.

In response to inquiries we are glad to be able, through the courtesy of Babcock and Wilcox, Ltd., to give the following particulars of the steam pipes at Langerbrugge power station. All joints in the high-pressure steam pipe line (800 pounds) are made as follows: The pipe flanges are faced off flat and square. Between them is inserted a disc of mild steel, machined all over, bored to the pipe diameter and turned to such a diameter that it touches the flange bolts. On each of its faces is a series of V-shaped con-



HIGH-PRESSURE STEAM PIPE JOINT.

centric grooves extending from the inner diameter of the disc to nearly the inner edges of the bolt holes. The working face of the disc thus consists of a number of sharp-edged ridges, the remainder being turned down to the thickness at the bottom of the grooves, and therefore leaving a clearance

space when the disc is in position. The bolt holes are reamed in position, turned and fitted bolts being used. Before the joint is made the grooves of the disc are smeared with a graphite paste, but that is possibly not essential. The flanges are screwed to the pipe, expanded, and welded. The form of the bolts should be observed.—"The Engineer," Aug. 6, 1926.

CHROMIUM ALLOYS RESIST CHEMICALS.

By C. E. MacQuigg.

The field of alloying presents almost unlimited possibilities [in resistance to corrosion] and is being rapidly developed. Metals, which are not by themselves immune to attack when alloyed in a proper manner, are quite satisfactorily resistant. The general principles of chemical stability are hardly more than glimpsed in our present knowledge of alloys, but some general considerations may be noticed.

GROUPING METALS AS TO RESISTANCE.

Metals (and alloys) may be divided into two main groups according to whether they owe their resistance to corroding influences to:

(1) low electrode potentials or solution pressures, or

(2) the formation of a protective coating, formed by the product of the corrosive action itself.

The first named effect will readily be recognized as due to the relative position of the material in the electrochemical series, as platinum, gold, silver, copper, lead, nickel, cobalt, iron, zinc, manganese, aluminum and magnesium, mentioned in the descending scale of general resistivity, or expressed differently, in their ascending tendency to go into solution in electrolytes .

The second effect will be a function of the nature of the film or product of the reaction. For example, although aluminum has a very high solution pressure, and thus an enormous tendency to go into solution, it is actually quite insoluble in nitric acid, because the very thin protective coating of more or less hydrated aluminum oxide, which is instantly formed, successfully resists further oxidation of the subjacent metal.

From the list of pure metals at present commercially feasible and having physical properties that make them amenable to fabrication, we must choose the following: iron, nickel, aluminum, copper, lead, zinc or tin.

RESISTIVITY A FUNCTION OF METALLOGRAPHIC PARTS.

With the somewhat restricted possibilities seen above, the outlook is much more promising in the direction of the alloy field. The resistivity of an alloy to corroding action is a function of the properties of its metallographic constituents; these may be either pure metals, solid solutions, definite chemical compounds, or their mixtures. The interrelation of chemical resistivity and structural constitution has not yet been the subject of more than a cursory investigation, but it seems safe to say that this study affords greater means of advance than any other, since more information of an indicative nature will be discovered by a correlation of metal structures with chemical properties, than any other means.

Alloys are loosely classified into ferrous and non-ferrous, depending on whether or not the base is metallic iron. While this classification was formerly more satisfactory, because formerly nothing but iron or steel had properties such as amenability to heat treatment and certain features of

microstructure, such is not now the case because some non-ferrous alloys have properties very similar to steel. Ferrous alloys are particularly subject to attack by acid or oxidizing agents. On the other hand they have those desirable attributes of workability and strength and, in addition, they are the cheapest.

Effort has been directed in the rather recent past toward alloying with iron some element or elements which would enhance its chemical stability without sacrifice of physical properties and undue increase in cost. Numerous elements were and are being tried, such as nickel, molybdenum, copper, cobalt, tungsten, titanium, manganese, vanadium, silicon, both singly and in combination. None has shown the results that have been obtained singly by chromium. Very desirable effects are also achieved with the chromium-nickel combination with iron, sometimes modified by silicon.

In the non-ferrous group almost unlimited possibilities exist for the discovery of corrosion resistant materials. The control by heat treatment of the various solid solutions and chemical compounds with their respective effects on physical and chemical properties all go to make probable the belief that here will occur in the next generation some outstanding achievements of metallurgy.

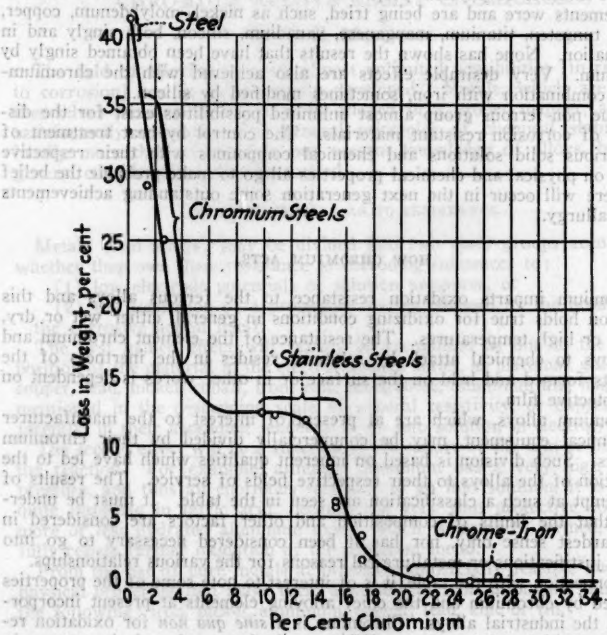
HOW CHROMIUM ACTS.

Chromium imparts oxidation resistance to the ferrous alloys and this condition holds true for oxidizing conditions in general, either wet or dry, at low or high temperatures. The resistance of the element chromium and its alloys to chemical attack by solutions resides in the inertness of the products formed and held on the surface or in other words is dependent on the protective film.

Chromium alloys, which are at present of interest to the manufacturer of chemical equipment, may be commercially divided by their chromium contents. Such division is based on inherent qualities which have led to the adaptation of the alloys to their respective fields of service. The results of an attempt at such a classification are seen in the table. It must be understood that the limits of composition and other factors are considered in the boardest sense only, nor has it been considered necessary to go into minute justifications or metallurgical reasons for the various relationships.

In connection with the table, it is of interest to note some of the properties imparted by chromium and the chief alloying elements at present incorporated in the industrial alloys. Chromium is a *sine qua non* for oxidation resistance at high temperatures. When used alone, unless it is present in amounts upward of at least 10 or 15 per cent, the alloy will not have the best chemical resistance. The oxidation resistant property of chromium is also imparted to its alloys against attack in the wet way, nitric acid being without effect if the metal contains about 12 per cent or more of chromium even in the absence of other alloying elements in appreciable amounts. If the element is used as the single alloying component, it will be found that the degree of immunity imparted is much higher if the chromium content is over about 20 per cent. This is particularly true when the material must withstand oxidation at high temperatures, as will be seen by the illustration, which shows the loss in oxidation plotted against chromium content, no other alloying element being present. It must also be pointed out that the oxidation loss shown by the high chromium alloys is about the maximum which would be obtained with indefinite exposure; this is not true for lower chromium contents since a critical point seems to exist around 20 per cent chromium.

For resistance to wet attack in other than oxidizing media, no general rules can be formulated. Chromium and chrome-nickel combinations have a remarkable versatility. Among the chemicals which are successfully withstood are acetic acid, many hydroxides, fruit and vegetable juices and mine waters. Against sulphuric acid the plain chromium iron alloys are not reliable under all conditions, although some installations have been quite successful. Unfortunately they are easily attacked by hydrochloric acid. Sulphur and sulphur gases are practically without effect on straight high chromium alloys. The presence of nickel, however, is disastrous, because of its vulnerability to sulphur.



Loss in Weight by Oxidation of Specimens of Iron with Increasing Chromium Content.

Chromium imparts high tensile strength to its alloys but, in the higher ranges of chromium content, somewhat decreases their ductility. Alloys containing 20 per cent chromium and upward are readily worked by the usual steel mill equipment to form plates, sheet, rods, wire, seamless tubing, etc. A peculiarly advantageous property of chromium is that of imparting high tensile strength at high temperatures; this opens up a field of applications which is becoming quite extensive. The presence of nickel greatly enhances the strength and stiffness and diminishes the elongation at high temperatures, thus increasing the resistance to flow under sustained loads at high temperatures.

TABLE OF COMMERCIAL CLASSIFICATION OF CHROMIUM ALLOYS.

Characteristic Chromium Range.	Principal Properties.	Some Typical Applications.
0 to 5 per cent Cr. With or without one or more alloying metals, as nickel, vanadium, molybdenum, etc.	High strength, ductility and toughness and (or) hardness.	Ball bearings, tools, armor, armor piercing shells, high speed cutting tools, automobile and airplane parts. Stamp shoes, crusher parts and rolls for ore preparation, grinding equipment, saws, files, multi-ply plates for safes and vault manufacture, etc.
14 to 16 per cent Cr. added to cast iron.	Resistance to oxidation where the physical properties of cast iron are satisfactory.	Certain types of oxidation resistance requirements where the application will not bear the expense of the higher quality alloys, as in annealing boxes for malleable cast iron foundries, roasting or muffle furnaces for temperatures up to 800 degrees C. (1475 degrees F.).
12 to 16 per cent Cr. Carbon content carefully controlled.	Resistance to corrosion together with excellent physical properties.	Cutlery, kitchen utensils, restaurant and hotel fittings, chemical plant apparatus to withstand nitric acid and very many other chemicals. Good against oxidation up to about 800 degrees C. (1472 degrees F.). Also being used extensively in engineering applications—as in turbine blades, internal combustion engine valves, etc. Builders' hardware and decorative parts, sporting goods, marine equipment, etc.
10 up to 20 per cent or more Cr. together with considerable percentage of other alloying metals, nickel being the most frequently used. Silicon and manganese are present in some analyses up to several per cent.	High electrical resistivity and non-oxidizing. High tensile strength at elevated temperatures is a feature. Some have hot and cold working properties.	Electrical Heating Elements. All kinds of heat-treating equipment such as case carbonizing boxes, annealing boxes, pyrometer tubes, furnace parts, etc.
20 per cent Cr. and upward. No appreciable alloying elements other than carbon.	Resistance to oxidation and to many forms of chemical attack, especially nitrates and sulphur. Softness and workability in low carbon ranges; hardness.	All types of apparatus which are subjected to oxidation up to about 1100 degrees C., such as stills, muffles, retorts. Crushing and grinding equipment. Nitric acid plant equipment. Mine pumps.

WELDING CHROMIUM ALLOYS.

A matter important to the chemical equipment manufacturer is the question of weldability of the chromium alloys. They do not hammer-weld but may be readily joined by fusion welding, either gas or electric. Because of the increasing commercial interest in the subject, some general observations will be in order.

The welding of chromium-iron alloys is attended with more difficulty than is the case with ordinary steels, due to the formation of infusible chromium containing oxides. The oxide formed is difficult to float and, unless eliminated, may give rise to such defects as blowholes, laps, and similar defects. The chromium-iron alloys may be satisfactorily arc welded, using for an electrode a wire of the same material coated with a flux which is capable of combining with or dissolving the oxides formed. Such electrodes, when suitably coated and used with the proper polarity, will give good dense welds. Some flux coatings will yield good results with either polarity, while others will work with reversed polarity only (electrode positive). Welds made by this process will possess good strength, are easily machinable, but will lack in ductility. For ordinary batch mixing or storage tanks, this lack of ductility in the weld would be of no consequence. However, in vessels destined for high pressure and temperature, lack of brittleness is important, and no such work should be undertaken without first consulting with someone experienced in the making and properties of welds in high chromium-iron alloys.

In order to oxy-acetylene weld the high chromium-iron alloys, it was formerly found that satisfactory welds could be obtained by using an excess of acetylene in the welding flame. While this procedure does prevent oxidation and makes the welding easy, at the same time it charges the weld metal with considerable carbon, making it quite brittle. This practice or method of welding, while satisfactory in some instances, is not generally recommended.

Neutral flame welding may be used by coating the line of the weld and the welding rod with special fluxes capable of protecting the hot metal and dissolving any oxides formed. Another solution to neutral flame welding is to use welding rods containing appreciable quantities of manganese and silicon whose oxidation products yield a fusible protecting slag. It is recommended that a small amount of flux be used on the line of the weld to take care of oxidation of the base metal. Research work is proceeding in this field and it is believed that the peculiar problems met in this type of welding will soon be overcome to permit safe welds for high pressure and resistance to shock.—"The Iron Age," Aug. 12, 1926.

PLATING WITH CHROMIUM FOR WEAR.

By C. H. Humphries.

Uses of chromium as a protective coating for various articles fabricated of iron and steel are multiplying daily. Recent application of this metal to commercial practice has effected savings in many industries totaling hundreds of thousands of dollars.

Particularly advantageous has been the chromium coating on rubber-tire molds. At the plant of an eastern tire manufacturer the current practice of tire making demands the cleaning of the tire molds after curing approximately 100 tires. Until recently, soap solution or soapstone was used regularly to prevent the tires from sticking in the molds and to preclude blem-

ishing of the finished tires. The soapstone, sprinkled from a filtering bag, built up deposits on the tire-mold walls. The deposits resulted after a certain time in making less pronounced the definition of the tread characters and the size and name lettering, so that the deposited skin finally had to be removed from the mold.

Another difficulty was the aggregation on the surface of the mold of sulphide of iron, which is produced by the reaction of sulphur from the rubber mix upon the iron of the surface of the mold. Its presence is indicated readily when hydrochloric acid, as a rinse or pickle, comes in contact with the black lining of the molds.

Adoption of chromium plated molds has eliminated the need of soapstone, and has shown that chromium as a surfacing for the inside of automobile tire molds prevents the formation of the troublesome skin previously mentioned, thereby saving tire manufacturers considerable money.

Under the old system tire molds are cleaned by burning away the rubber deposits with an oxy-acetylene flame, and by mechanically removing the residue by means of steel wool or scratch brushes. Steel or cast iron molds cost from $90 to $600, depending on their size and design. On an average, approximately 10 per cent of the molds are idle for the cleaning operation, which is neither short nor easy. It is obvious that a tire plant having 6000 to 8000 tire molds as part of its equipment would have considerable investment idle, with 10 per cent of the molds out of production during the cleaning period.

Only a relatively small outlay is required for equipment sufficient to process with chromium all of the molds in use as well as the new ones. Furthermore, the actual cost of chromium plating is inconsiderable. The maintenance cost for each mold when processed, as compared with the cost of the frequent cleaning otherwise necessary, represents a saving of from $25 to $100, according to the composition of the mixture, the condition of the cure, etc. In this particular field there are many other uses for chromium plating, because tire molds make up but one class of many molds utilized for fabricating rubber articles.

REFLECTOR LININGS FOR LONG LIFE.

Another interesting application of deposited chromium coating is as a lining for reflectors. The present method of lining calls for silver plating direct, or silver plating over nickel and then buffing to a high, lustrous finish. Though silver has a high light reflective value, its service is not permanent because of discoloration, chiefly through the formation of black sulphide of silver.

Thin silver mirrors are scratched easily and their reflective value is lessened. Chromium, on the other hand, can be deposited on a highly finished reflector surface in a condition of high luster, which does not require buffing. It neither tarnishes nor discolors, and is so hard when deposited in the right physical condition that it does not scratch when cleaned, even if ashes or soot are on its surface.

A comparison of the silver lined mirrors or reflectors with those which have been chromium coated reveals that the former has a slightly higher initial luminous value, approximately 95 per cent as against 90 per cent for the latter. This is due to selectivity, because of the color of a buffed chromium surface. After a few weeks of exposure, however, the ratio of reflective value is changed, with chromium standing at 75 per cent and silver at 40 to 60 per cent.

The average locomotive reflector has over 1 ounce and sometimes up to 3 ounces of silver deposited as a mirror, which must be replaced almost yearly. Therefore, substitution of a permanent reflective surface, such as chromium, is desirable from the standpoint both of cost and of efficiency. The average time for depositing a chromium mirror is less than that for depositing a silver mirror.

PLUMBING FIXTURES FOR HARD WEAR.

Application of chromium coverings to plumbing fixtures has proved practical. The shoddy appearance of bathroom fixtures from which the nickel coating has worn is offensive to the eye. It is caused by abrasion on the used parts, such, for example, as on the handles of the faucets. When chromium is substituted for nickel the color resembles platinum, because of its bluish tint. Because of the difference in hardness, the chromium treated surface will outlast the nickel appreciably. Standard nickel plating can be cut through with little effort during the buffing operation, but thinner coats of chromium withstand efforts to cut through them even with sharper buffing compositions and bigger and faster buffing wheels.

Nickel plating is only fairly continuous, in that it has a number of capillaries per unit of surface. At the bottom of the capillaries moisture, carbon dioxide and ammonia cause local action which produces verdigris or a disagreeable green film on the surface of the fixtures, which in time causes the nickel to lift off. Chromium, on the other hand, has few pores and is unique in preventing the formation of this troublesome growth. Therefore, the use of chromium in the plumbing field has effected material savings and has enhanced the beauty of fixtures. A practical example is the employment of chromium as a coating for automatic coin boxes in wash rooms and toilets.

Among other products using chromium as a protective cover are automobile lamp bodies, automobile bumpers, hardware, surgical instruments, door kick plates, serving trays, office and stair railings, chandelier and wall fixtures, precision tools, and various kinds of novelties. The list includes articles for which the top coat is considered not only a finish, but also a protection against corroding media. It can be augmented greatly when cases of protection against specially destructive evils are included.

HEAT-OXIDATION WELL RESISTED.

Resistance to oxidation from heat is a valuable property of chromium. As a protector for oven linings it is effective to a marked degree. There is no raising or checking of the coat and no discoloration or tinting below 350 degrees C. (660 degrees F.). Even for protection against scaling at higher temperatures it is satisfactory, as it does not oxidize readily under 1100 degrees C. (2000 degrees F.). Valve stems of internal combustion engines and steel pyrometer tubes covered with chromium can be heated and cooled repeatedly without scaling, and under extreme conditions stand up well.

Processing with chromium the individual exhaust ports of airplanes affords a real test. Without a protective coating they show red rust and, although they operate at a red heat, when covered with chromium the only effect is a brown or blue tinting, with no chipping or scaling.

Common electric irons or sad irons remain smooth after a coating of chromium. In fact, there is less tendency for the starch to stick. Even special articles, such as the heating units used for the permanent waving of women's hair, have been protected against oxidation by heat by means of chromium plating.

HARDNESS TO RESIST WEAR.

Chromium deposits are extremely hard. The hardness can be controlled and varied between 60 and 85 on the scleroscope. The hardness of the coats gives protection to surfaces in which resistance to abrasion or wear is an important factor. A film of chromium less than 0.0003 inch thick increases about six times the number of prints which can be taken from a copper electrotype.

This resistance to wear makes possible the elimination of some hardening operations on steel wearing parts. On crankshafts and cams in internal combustion engines the chromium coat gives sufficiently long life to make it unnecessary to harden to the glass hard and consequently to the embrittling point.

Early in 1923 the writer chromium-processed some golf clubs which were distributed for test purposes by a well-known sporting goods manufacturer. They have been used under severe conditions, but the scratches from sand traps and gravel are slight. They have not rusted and have not lost their original luster.

Obviously it is impossible to relate all of the applications of chromium. Its resistance to fruit juices and all common acids, except hydrochloric and sulphuric, has made its use on fruit juice concentrators, tomato kettles and concentration coils almost imperative. Acetic and citric acids, found in so many food products, show no action on chromium-treated linings.

Alloys of chromium with cobalt are still more resistant, and parallel the properties of the alloy Stellite. Work with this alloy dates back seven years and the promises of its performance in certain fields exceed those of pure chromium.—"The Iron Age," Sept. 2, 1926.

BOOK REVIEWS.

MOTORSHIP YEARBOOK—4TH EDITION, 208 PP., PRICE $3.00. PUBLISHED BY MOTORSHIP, 220 W. 42ND STREET, NEW YORK CITY.

The fourth edition of Motorship Yearbook shows vividly the growth of the progress in motorships since the first modest experiment in 1904.

Forty-five pages are devoted to a list of some 900 vessels which have machinery exceeding 300 H. P. In this list appear passenger vessels, freighters, tankers, tugs, ferries and dredges.

The application of the Diesel engines to all types of vessels is described as well as the modern aids to safe navigation.

A list of radio beacons of the world is given as well as one of oil bunkering stations.

The book is an up-to-the-minute compendium of information for those desirious of keeping posted on the latest practice in motorships.

ASSOCIATION NOTES.

The annual business meeting of the Society was held on Tuesday, October 5, 1926 at Washington, D. C. The following nominations were made for officers of the Society for the year 1927:

For *President:*

Rear Admiral John Halligan, Jr., U. S. N.

For *Secretary-Treasurer:*

Commander H. B. Hird, U. S. N.

For *Members of Council:*

Rear Admiral C. W. Dyson, U. S. N.
Captain E. L. Bennett, U. S. N.
Captain L. B. McBride, (CC) U. S. N.
Captain Q. B. Newman, U. S. C. G.
Captain S. M. Robinson, U. S. N.
Captain Carlos Bean, U. S. N.
Commander H. H. Norton, U. S. N.
Lieutenant Commander E. L. Cochrane (CC) U. S. N.
Mr. J. F. Metten.
Mr. H. M. Southgate.

Ballots have been mailed to all members entitled to vote.

It was voted to hold the usual annual banquet in 1927.

At a meeting of the Council of the Society the resignation of Commander A. M. Charlton as Secretary-Treasurer was accepted to take effect on November 15, 1926 due to his detachment from duty in Washington. In accordance with the provisions of the by-laws of the Society, Commander H. B. Hird was appointed by the Council to serve the unexpired term of Commander Charlton.

The following members have joined the Society since the publication of the August JOURNAL:

NAVAL.

Draim, Nicholas, Lieutenant (CC) U. S. N.
Hinckley, Walter F., Lieutenant, U. S. N.
Kaster, Howard B., Ensign, U. S. N. R.
Kreinbihl, Louis J., Chief Machinist, U. S. N.

CIVIL.

Barrett, J. M., 558 West Fremont St., Fostoria, Ohio.

ASSOCIATE.

Ciarlo, Esteban, Capt. Engineer, Argentine Navy.
Coe, Frank C., 316 Bulletin Bldg., Philadelphia, Pa.